FINAL REPORT

PREPARED FOR

UNITED STATES MARINE CORPS

DEVELOPMENTAL TEST REPORT
FOR
LOGISTICAL VEHICLE SYSTEM REPLACEMENT (LVSR) PROGRAM

AUTHORIZATION:
CONTRACT NUMBER: N00167-98-C-0017

NATC PROJECT NUMBER 20238

JULY 2001 – JANUARY 2002

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1.0 BACKGROUND

The United States Marine Corps contracted the Nevada Automotive Test Center (NATC), under Contract Number N00167-98-C-0017, to conduct Developmental Testing (DT) of the Logistical Vehicle System Replacement Technology Demonstrator (LVSR-TD) vehicle. As there was only one LVSR-TD vehicle built, the testing occurred in and around other USMC programmatic demonstrations/ shows, the USMC Early Operational Assessment (EOA), and the USMC Measures of Performance(MOP) testing for the Analysis of Alternatives (AoA) study. The overall developmental test schedule is shown in Table 1. For the ride quality and stability testing, an air ride and a hydraulic suspension Rear Body Unit (RBU) was tested with the LVSR-TD Front Power Unit (FPU).

A complete description of the LVSR-TD is contained in the following additional reports and the reader is referred to these reports for additional information on the truck design and build. This report discusses the DT and results only.

[1] NATC, “Analysis of Alternatives for the Logistics Vehicle System Replacement (LVSR) Task 1, Interim Report, Literature Review,” NATC Project Number 17859, Prepared for United States Marine Corps, 6 June 2000.

[2] NATC, “Analysis of Alternatives for the Logistic Vehicle System Replacement (LVSR) Task 2, Interim Report, Description of the LVSR R&D Alternative Truck,” NATC Project Number 17859, Prepared for United States Marine Corps, 30 June 2000.

[3] NATC, “Analysis of Alternatives for the Logistics Vehicle System Replacement (LVSR) Task 3, Country Terrain Analysis,” NATC Project Number 17859, Prepared for United States Marine Corps, 30 June 2000.

[4] NATC, “Analysis of Alternatives for the Logistics Vehicle System Replacement (LVSR), Task 4, Calculate the Measures of Performance,” NATC Project Number 17859, Prepared for the United States Marine Corps, 30 April 2001.

[5] NATC, “Analysis of Alternatives for the Logistics Vehicle System Replacement (LVSR), Final Report”, NATC Project Number 17859, Prepared for the United States Marine Corps, 30 December 2002.

[6] Belcher, Gerald J, Bobby Jackson, Colin Ashmore, Muluneh Sime, “Analysis of Alternatives for the USMC Logistics Vehicle System Replacement, Final Report” LMI Report MC004T2, April 2001.

[7] Belcher, Gerald J, Bobby Jackson, Colin Ashmore, Muluneh Sime, “Modeling and Analysis for the Logistics Vehicle System Replacement Analysis of Alternatives: Interim Report #1,” LMI Report MC004T1, August 2000.

[8] Logistics Vehicle System Replacement (LVSR) Prototype Performance Specification (DRAFT).

[9] Operational Requirements Document for Logistics Vehicle System Replacement (DRAFT).

[10] NATC, “Logistics Vehicle System Replacement (LVSR) Safety Assessment Report,” NATC Project Number 17859, Prepared for the United States Marine Corps, 6 July 2000.

[11] NATC, Logistics Vehicle System (LVS) Signature Data”, NATC Project Number 20513, Prepared for the United States Marine Corps, 30 January 2003.

[12] NATC, “Summary Report LVS Mod Demo Vehicle Armor Protection Kit Installation”, NATC Project Number 17859, Prepared for Naval Surface Warfare Center, 30 August 2001.

Table 1
LVSR Schedule of Events After Vehicle Build and Checkout

EOA – Early Operational Assessment – Operator’s Portion	July 24 – August 2, 2000
EOA – Early Operational Assessment – Maintainer’s Portion	August 3 – August 11, 2000
Scheduled maintenance of vehicles after EOA	August and September, 2000
LVSR-TD, Raydan and Hendrickson performance, ride quality, stability and model validation testing	October 2000 – February 2001
Fuel Economy	Late February – Early March 2001
Environmental Chamber, Transportation Symposium, Camp Pendleton, CA and Congressional Day, Quantico, VA	March – June 2001
Durability/RAM-D Testing	Late June to August 2001
Instrumentation for Full Load Cooling	Late August 2001
Full Load Cooling Test	Late August - September 2001
Prepare vehicle for shipment to USMC Quantico facility for Modern Day Marine Expo	Early September 2001, Modern Day Marine cancelled
Prepare vehicle for shipment to Army AUSA show	Early October 2001, AUSA show cancelled
USMC Personnel Training at NATC and Ship LVSR-TD to Quantico, VA	November 2001

2.0 EXECUTIVE SUMMARY

The LVSR-TD supported the development and refinement of the draft Operational Requirements Document (ORD) and the draft Performance Specification for the LVSR program. Much of the LVSR developmental testing data are contained within these documents. This report presents the results of the major LVSR developmental tests and performance measurements. The LVSR met the performance requirements of the draft Performance Specification as summarized in this report.

3.0 OBJECTIVES

The objectives of these tests were to verify that the LVSR meets critical technical parameters outlined in the draft ORD for the LVSR program. A second objective was to assist the program in validating the draft Performance Specification requirements.

The LVSR DT consisted of the following series of tests:

- 2,000 mile accelerated RAM-D test
- Limited FMVSS 121V brake certification test
- Environmental chamber cold room test
- Full load cooling
- Ride quality
- Fuel economy
- Performance testing
- Stability and handling

4.0 TIRE PRESSURES

The tire pressures in Table 2 were used throughout the LVSR-TD testing. These pressures were also programmed into the Central Tire Inflation System (CTIS) controller. When the alternate Raydan and Hendrickson RBUs were tested, these same tire inflation pressures were used.

Table 2
Tire Pressures for the LVSR-TD Testing

		Full Payload (22.5 Tons)		Half Payload (18 Tons)		Empty	
Terrain	**Speed**	Axles #1/2 7,250 lb/tire (PSI)	Axles #3-5 10,787 lb/tire (PSI)	Axles #1/2 7,323 lb/tire (PSI)	Axles #3-5 9,620 lb/tire (PSI)	Axles #1/2 6,780 lb/tire (PSI)	Axles #3-5 4,133 lb/tire (PSI)
Highway	65	50	84	50	68	50	30
Cross-Country	45	26	58	26	47	26	19
Multi-Terrain	30	14	32	14	25	14	12
Snow/Ice	25	14	32	14	25	14	12
Deep Mud	20	14	32	14	25	14	12
Sand	20	13	23	13	20	13	9

The only exception to the above tire pressures were a slight variation for the tilt table stability testing. The tire pressures were increased to reduce the tire deflection during the tilt table evaluations (Table 3).

Table 3
Tire Pressures for the LVSR-TD Tilt Table Testing

Half Payload (18 Tons)		
Terrain	Axles #1/2 7,323 lb/tire (PSI)	Axles #3-5 9,620 lb/tire (PSI)
Highway	60	90
Cross-Country	48	62

5.0 VEHICLE WEIGHTS

Tables 4 through 6 show axle-by-axle weight distribution for the LVSR-TD at curb weight and two payloaded weight configurations. The Cross-Country Gross Vehicle Weight (CCGVW) payload was 18 tons (Table 5). The Highway Gross Vehicle Weight (HGVW) payload was 22 tons (Table 6). In Tables 4 through 6, axles 1 through 5 are front to rear axles, respectively. The weights were designed to best simulate a 16.5 ton payload on a 1.5 ton M1077 flatrack (CCGVW) and an ISO container weighing a total of 22 tons (HGVW). The LVSR-TD had the current MK-18A1 Load Handling System (LHS) installed and no changes were made to the LHS. Payload adjustments were made for the Front Lift Adapter (FLA) as it was assumed that an FLA would not be required on the final design of the LVSR-TD and intelligent LHS.

Table 4
LVSR-TD Curb Weight, Full Fuel, Full BII, No Operator/Crew
(This includes the FLA connected to the MK-18A1 hookarm)

AXLE	LEFT (Pounds)	RIGHT (Pounds)	TOTAL (Pounds)
1	7,090	6,830	13,920
2	6,900	6,300	13,200
3	5,470	5,120	10,590
4	3,880	3,900	7,780
5	3,650	2,780	6,430
TOTAL	26,990	24,930	51,920

Table 5
LVSR-TD 18-Ton Payload, Full Fuel, Full BII, No Operator/Crew
(Payload was an ISO Container Payloaded to Simulate 16.5 Tons on a 1.5 Ton Flatrack - 24 inch high CG)

AXLE	LEFT (Pounds)	RIGHT (Pounds)	TOTAL (Pounds)
1	7,380	7,160	14,540
2	7,330	7,420	14,750
3	9,510	9,820	19,330
4	9,440	9,800	19,240
5	10,010	9,140	19,150
TOTAL	43,670	43,340	87,010

Table 6
LVSR-TD 22-Ton Payload, Full Fuel, Full BII, No Operator/Crew
(Payload was an ISO Container Payloaded to 22 Tons - 48 inch high CG)

AXLE	LEFT (Pounds)	RIGHT (Pounds)	TOTAL (Pounds)
1	7,070	7,090	14,160
2	7,610	7,230	14,840
3	10,810	10,480	21,290
4	10,510	10,900	21,410
5	11,730	10,290	22,020
TOTAL	47,730	45,990	93,720

6.0 FUEL AND LUBRICANT

Throughout the testing, performance tests were run on diesel fuel (DF-2) and the RAM-D test was run on JP-8 fuel. The one exception was the cold room performance test, which was run on JP-8 in lieu of switching to an arctic grade fuel. Table 7 summarizes the fuel type used for the various tests. Additionally, ExxonMobile Delvac 1 was used in the single lubricant, single reservoir system throughout the testing.

Table 7
Summary of Fuel Used For LVSR-TD Testing

Test	Fuel
EOA/AoA Testing	Diesel (DF-2)
Ride Quality	Diesel (DF-2)
Stability	Diesel (DF-2)
Acceleration	Diesel (DF-2)
Maximum Speed	Diesel (DF-2)
Full Load Cooling	Diesel (DF-2)
Cold Room	JP-8
Accelerated RAM-D	JP-8

No anomalies occurred using either the JP-8 fuel or the synthetic Delvac 1 synthetic oil. The powertrain design of the LVSR-TD validated the performance requirements related to reduced logistic footprint. The use of a single fuel and single lubricant was validated.

7.0 RAM-D TEST

An accelerated reliability, availability, maintainability and durability (RAM-D) durability test was conducted on the LVSR-TD from June to August 2001. The 2,000 mile accelerated RAM-D test was designed to determine the predicted system reliability of the LVSR-TD against the established mission profile. The purpose of the accelerated durability test protocol was to determine the vehicle's response to the rough off-road terrains and other environments to which the vehicle would be exposed.

Primarily, the design of the accelerated RAM-D evaluation was to determine the capability of the LVSR-TD to meet the performance and reliability objectives when operated for over 20,000 miles of the LVSR 40/30/20/10 mission profile. The durability testing was conducted over a range of test course roughness levels that accelerated the fatigue portion of the mission profile 10:1 (i.e., 2,000 test miles equals 20,000 mission profile miles).

This accelerated test design led to a test scenario more closely aligned with the MTVR mission profile of 10/20/30/40. However, no course was rougher than the mission profile of the LVSR-TD within the USMC scenarios.

Unofficially, the LVSR-TD had approximately 13,000 miles accumulated on the vehicle before the RAM-D testing was initiated. This mileage was mostly off-highway operation in support of the EOA, AoA, numerous demonstrations, ride-quality and performance testing. Although not part of the RAM-D evaluation, this mileage assisted in the validation of the powertrain and rotational components on the LVSR-TD.

7.1 RAM-D Test Specifications

The “Durability” paragraph of the draft Performance Specification states "Each variant shall demonstrate no major component durability failure during its first 20,000 miles of operation. Major components include the engine, transmission, transfer assembly (if used), frame, axle assemblies, and load handling system. A durability failure is defined as the need for replacement or overhaul of a major component”.

The “Duty Cycle/Mission Profile” paragraph of the draft Performance Specification states "The following definition describes the LVSR duty cycle/mission profile. Unless otherwise specified, performance shall be demonstrated on surfaces such that 40% is completed on Improved Hard Surface Roads, 30% on Improved Gravel Roads, 20% on Unimproved Surface Trails, and 10% Unimproved Surface Cross-Country. The Government has defined duty cycle/mission profile percentages and RMS values for surface roughness”.

The “Reliability” paragraph of the draft ORD and Performance Specification states “Each variant shall demonstrate a minimum Mean Miles Between Operational Mission Failure (MMBOMF) reliability of not less than 4,000 miles of operation (threshold), 6,000 miles (objective)”.

7.2 RAM-D Initial Inspection

Prior to RAM-D test initiation, the LVSR-TD received a detailed initial inspection. As the RAM-D was one of the last tests completed on the LVSR-TD, the inspection was used to identify any conditions that would require repair or replacement prior to test initiation. As part of the initial inspection, the vehicle was re-weighed to verify the axle/wheel loading at curb vehicle weight (CVW) and at CCGVW.

During the inspection, no major anomalies were noted. The number one and five axles had blown CTIS seals and the front CTIS controller was replaced.

7.3 RAM-D Inspections

During RAM-D shift operations, the test vehicle received pre-shift and post-shift inspections by the vehicle operator/maintenance technician. After every shift, the vehicle received a bumper-to-bumper inspection by a test vehicle maintenance technician and engineering support staff. During this inspection, any repairs or components that required replacement were documented. All of the incidents observed by the test vehicle operator/maintenance technician and test engineer were documented in the test vehicle log. A representative copy of an operator’s driver log documenting one shift of operations is provided in Appendix A.

When conditions that might require test vehicle modifications were observed, the USMC was notified. Any changes to vehicle configuration due to developments during the durability operations were documented.

7.4 RAM-D Vehicle Weight Configuration

One hundred percent of the RAM-D testing was performed with the LVSR-TD at the CCGVW in Section 5.0 at the tire pressures specified in Section 4.0 for 18 tons.

7.5 RAM-D Test Procedures

7.5.1 RAM-D Instrumentation

Prior to test initiation, the LVSR-TD was equipped with a NATC designed on-board data acquisition system called the Solid State Vehicle Recorder (SSVR). Accelerometers were installed at the left front axle, left rear axle and the base of the driver's seat. The SSVR recorded vehicle speed and accelerations during RAM-D operations. A representative copy of an SSVR printout documenting one shift of operations is provided in Appendix A.

7.5.2 RAM-D Test Operations

One shift of 10 hours each was scheduled Monday through Friday. The shift was scheduled to depart at 0800 hours. Between 0600 and 0800 the vehicle was inspected (PMCS), fueled and maintained as required. Any anomalies noted by the test vehicle operator/maintenance technician were annotated into the vehicle's logbook.

The test course was structured so that each shift of operation would include representative segments of terrain specified in the LVSR mission profile, except for paved road. A Test Engineer performed baseline test operations. The purpose was to profile the test course for segment miles, times, and appropriate safe and representative operating speeds for each segment. Included in this data were gear positions, differential modes, segment times, maximum and average speeds, CTIS settings and other pertinent information for safe and consistent test course operations. Test operation profiles were established in the CCGVW mode.

The data from these profiles were used to prepare Route Logs for the test vehicle operator. The operator was instructed to operate the vehicle in accordance with the details of the route log. A copy of the route log has been provided in Appendix D. At each significant terrain change, the driver was required to record the clock time and odometer reading. These logs, combined with the data from the SSVR, were used to monitor the driver's performance and provide the Test Engineer with test miles and average speeds on each terrain type. The odometer in the test vehicle was used as "event markers" for beginning and end of shift, test incidents, etc. However, all test miles accumulated were based on the actual miles accumulated under the route and driver's log format as processed. Slight differences between the accumulated test miles and the odometer miles occurred due to operation of the LVSR-TD for non-test events or demonstrations.

Each shift of test operation for the vehicle was documented with a driver's log and SSVR chart. Route deviations, where required, were documented as to route, time, and miles on the driver's log. Driver's logs and SSVR charts were processed on a daily basis. The processed information was posted to computer spreadsheet master logs as well as logs detailing the miles and times accumulated by course type. A summary of the test mileage and terrain type is listed in Appendix E.

Any equipment malfunctions that occurred during testing were recorded in the test log. In accordance with USMC instructions, TIRs were not required for the LVSR RAM-D testing. Documentation of failed components, systems included:

- Component/system nomenclature and part number (if known)
- Test miles on component/system

- Seriousness of failure regarding down time, mission completion, etc.
- Reason for failure and circumstances under which the failure occurred
- Corrective action taken
- Recommendations for preventative maintenance
- Location of failed component

7.5.3 RAM-D Test Courses

The test director determined the mix of courses to equate to a 10:1 acceleration factor. Profiles of the test course for segment mileage, times and appropriate safe speeds for each segment were developed. Course speeds were determined based on subjective evaluation of ride quality and performance.

The course was developed to match the terrain/mission scenario mix listed in the LVSR draft Performance Specification. The accelerated test included portions of the following NATC courses (Table 8).

Table 8
NATC Test Courses Used For RAM-D Testing

Road Name/Type	NATC Course Name
Gravel	Access Roads
	Gravel Oval
Perryman I	Access Roads
	Perryman III to North Butte
	Shop to Tank Course (River Crossing)
	Churchill Canyon
	Forest Service Loop
Perryman II	Adrian Valley
Perryman III	Perryman III
Belgian Block	Belgian Block
Sand	Sandwash I – III
	Sand Serpentine
Churchville B	Tank Course
	Bull Canyon
	North Butte
	Battlefield Loop
	Susan’s Bluff

7.5.4 RAM-D Test Results

The RAM-D testing began on 26 June 2001. The odometer reading in the vehicle was 2,982.5 at the onset of the test. The photographs for the RAM-D test are contained in Appendix B, Photographs 20238-001 through 20238-008.

The vehicle completed 2,001.6 test miles with an 18 ton payload consisting of a twenty-foot ISO container.

The LVSR-TD met the durability and reliability requirements of the draft LVSR Performance Specification. During test conduct, no mission failures occurred. The LVSR-TD incidents during RAM-D testing were as follows. Appendix F contains a full maintenance log for the RAM-D test.

- Failed #4 to #5 axle driveline. U-joint determined bad.

- Replaced output yoke (#4 axle), U-joint and rear slip yoke. The rear slip yoke cracked when it hit the ground during replacement.

- Eight Class 1 to Class 3 leaks. Most were at the transmission prototype valve body stackups and rear output shaft housing that were known non-production intent items.

- Thirteen flat tires - function of the test course roughness, not of the vehicle design.

- Failed two shocks on FPU and replaced two others due to lack of damping. No RBU shock failures. NATC expected to replace front shocks as these were MTVR shocks and light on dampening for the LVSR axle weights.

- Failed solid state relay in CAPSTART system.

- Cracked radiator hose – rewelded.

- Loosening of steering gear box at #2 axle (retorqued numerous times). It was replaced at the end of test.

7.6 RAM-D Conclusions

The LVSR-TD met the durability and reliability requirements in accordance with the draft Performance Specification and draft ORD.

8.0 BRAKE APPLY AND RELEASE TIMING AND STOPPING DISTANCE RESULTS

The LVSR-TD was subjected to a partial FMVSS 121V braking evaluation to evaluate stopping distances and apply and release timing. As part of the test, apply and release times, stability and control, and straight line stopping evaluations were performed. All portions of the test were performed with the LVSR-TD fully loaded with its HGVW rating (22 tons).

The LVSR-TD met the stopping distance and parking brake requirements at 22 tons (Table 9). The LVSR-TD met the apply/release timing requirements of FMVSS 121. Axles number 1 and 4 were instrumented and measured for apply and release timing, as it was known that axles 2 and 3 would be between these times based on air line routing.

Table 9
Brake Test Apply and Release and Stopping Distance Data

Apply and Release Timing (Seconds)

	FMVSS 121 Requirement (mSec)	LVSR-TD #1 Axle (mSec)	LVSR-TD #4 Axle (mSec)
Apply	0.50	0.268	0.406
Release	1.00	0.299	0.543

Stopping Distance (Feet)

Speed	FMVSS 121 Requirement (ft.)	LVSR-TD Avg Distance(ft.)
30 MPH	89	81.0
35 MPH	121	101.5
40 MPH	158	129.5
60 MPH	179	148.5

The stability and control evaluation was conducted on a 500 foot radius wet asphalt surface. This portion of the braking evaluations was performed on the low friction coefficient surface, each run increasing in speed and ending at 27.5 MPH. All brake stops were within the 12 foot lane without tire lockup.

The LVSR-TD parking brakes held the vehicle on a 40% grade at the HGVW.

9.0 ENVIRONMENTAL EVALUATIONS

The LVSR-TD was subjected to a cold room environmental chamber evaluation. A functional test was performed after the vehicle stabilized at -25 degrees F (Reference Photograph Numbers: 20238-009 through 20238-013). The cold room evaluation was performed at –25 degrees F. A scheduled 125 degree F hot room evaluation was delayed and time did not allow it to be rescheduled. During this test, the LVSR-TD was evaluated to ensure that all functions operated properly.

9.1 Environmental Test Objective

The objective of this test was to determine the limits of the vehicle as a function of extreme temperature.

9.2 Environmental Test Procedure

In preparation for the test, the engine oil was not changed as the vehicle ran synthetic Delvac 1 oil during the entire performance testing. The fuel was changed to JP-8.

Storage temperatures can be substantially below the standard ambient conditions. These temperatures can adversely affect seals and materials. In accordance with MIL-STD-810F, Method 502.4, the vehicle was exposed for a 24-hour soaking cycle at -25 degrees F. Low temperature startup operational tests were performed at –25 degrees F.

When the vehicle stabilized at the required temperature conditions, the operating evaluations were initiated. The operational integrity of the powertrain, brakes, engine, electrical components, etc. was analyzed for evidence of malfunction, performance degradation and/or failure at the low temperature.

9.3 Environmental Instrumentation

The instrumentation utilized for the environmental chamber evaluation consisted of the channels in Table 10.

Table 10
Environmental Chamber Instrumentation

Channel No.	Location	Units
1	Chamber Temperature	Deg F
2	Chamber Relative Humidity	% RH
3	Chamber Relative Humidity	% RH
4	Chamber Temperature	Deg F
5	Hydraulic Oil Temperature	Deg F
6	Radiator Temperature	Deg F
7	Rear Differential Temperature	Deg F
8	Front Differential Temperature	Deg F
9	Transmission Sump Temperature	Deg F
10	Oil Sump Temperature	Deg F
11	Coolant Out Temperature	Deg F
12	Coolant In Temperature	Deg F
13	Fuel Temperature	Deg F
14	Capacitor Amperage	Amps
15	Capacitor Voltage	Volts
16	Alternator Voltage	Volts
17	Alternator Amperage	Amps

The sampling rate was set at one sample every five minutes during the vehicle soak. While operating the vehicle, the data was collected at one sample/second.

9.4 Environmental Chamber Test Conduct/Results

The environmental chamber test was conducted in March and April 2001. The above instrumentation was installed. The first –25 degree F cold test was started on 29 March after a 24-hour cold soak. The first attempt at the –25 degree F cold start resulted in a dual failure of the starter, the starter solenoid and a solid state relay in the LVSR-TD prototype capacitive starting (CAPSTART) system.

It was determined that the solid state relay failure was due to either a failure in the starter solenoid or in the CAPSTART diode assembly. The starter was stuck in the engaged position, causing high temperatures on the wires, assemblies and solid state relays in the system. A CAPSTART diode assembly also failed during the course of this component failure. The relay and a diode were both replaced. The diode was replaced with a larger component. Both relay and diode operated normally without further incidence for the rest of the LVSR-TD testing.

After replacement of the starter and CAPSTART solid state relay and diode components, the LVSR-TD was transported to the USMC Transportation Symposium in Camp Pendleton, California from April 9-11, 2001. Upon the return from the Transportation Symposium, the vehicle was re-instrumented for the cold room testing. A scheduled 125 degree F hot room evaluation was delayed so that the –25 degree F cold room test could be repeated. After the second cold room testing, the LVSR-TD was transported to Quantico, VA for a Congressional Day demonstration.

The second –25 degree F cold test was started on 18 April after a 24-hour cold soak. During the first attempt at –25 degree F, the engine attempted to start, ran for five seconds and then died. The CAPSTART system was connected through a NATO slave cable to recharge the capacitors to 28 volts. The NATO slave cable was connected to a 200-amp supply from a HMMWV parked outside the environmental chamber. Upon the second attempt, the engine started and the LVSR-TD operational checks were initiated.

During the –25 degree F operations, it was found that the charging system was not holding the capacitor voltage at 28 volts. It was determined that the alternator regulator had a safety shutdown feature designed to protect the charging system and associated wiring when a short circuit or excessive current conditions exist. This issue caused the no charge condition (i.e. the alternator not sourcing current and recharging the capacitors). The manufacturer was contacted and a new 28V regulator was designed and built which addressed this issue. This alternator was never installed and retested at the –25 degree F condition.

The LVSR-TD incorporates flat panel displays that were part of the environmental evaluation at –25 degrees F. It was found that the flat panel displays came online with initial power on but went off-line while the vehicle was cranking. Upon starting, the flat panel displays functioned normally. However, it was found that the flat panel displays dropped out at 17 volts when the charging system was not able to maintain the required charging voltage. This was a feature of the CAPSTART system which shuts down less critical loads to prioritize power to the mission critical components. This CAPSTART load shedding mechanism operated normally in the environmental chamber. All other operational checks were performed with the NATO slave cable connected between the HMMWV and the LVSR-TD to supply the required 28 volts.

This charging anomaly precluded NATC from evaluating and quantifying the rapid warm-up feature of the LVSR-TD single lubricant, single reservoir system combined with the hydrostatic retarder. NATC attempted to validate this rapid warm-up feature on 19 April after another 20 hours of cold soak at –25 degrees F but was unsuccessful due to the alternator anomaly.

9.5 Environmental Chamber Conclusion

Based on the above results, it was concluded that the vehicle performed as intended at the low temperature operating conditions given the installation of a modified alternator and the redesign of the CAPSTART relay and diode. The hot room evaluation was not conducted due to scheduling conflicts and the higher priority test requirements to conduct RAM-D and full load cooling testing before the LVSR-TD was shipped to Quantico, VA.

10.0 FULL LOAD COOLING

The full load cooling evaluation was performed to determine if the LVSR-TD was in compliance with the “Cooling System” paragraph of the draft Performance Specification, which states that the cooling system shall be capable of limiting all vehicle fluids to the operating temperatures recommended by the component manufacturers while operating in the vehicle’s operating temperature range under the following conditions:

1. Continuous tractive effort equal to 60% of CCGVW at ambient air temperature of 125°F.
2. Any other road load condition within the mission profile to include improved road payloads.

The LVSR-TD CCGVW was specified at the design target of 82,000 pounds. Therefore, the net 0.6 tractive effort value would be 49,200 pounds of drawbar pull. Three tests were run at tractive effort values of 10%, 30%, and 60%.

10.1 Full Load Cooling Criteria

Full load cooling was performed in incremental steps up to the maximum tractive effort specified. Test operation was at wide open throttle against a mobile dynamometer (s). The lowest gear available was selected on a level hard surface with a coefficient of tractive effort high enough to sustain the anticipated drawbar values without gross tire slip.

Testing was not to be conducted if continuous wind velocity exceeded 7 MPH. Testing was conducted in greater than 70 degree F ambient temperature (due to the higher solar load and the smaller delta between the actual temperature and the values to which the temperatures were to be corrected). A one-to-one correction in temperature was used to correct the data to a 125 degree F ambient.

The test operation at the respective tractive effort value was to continue until each component or fluid temperature reached stabilization. A fluid/component was considered stabilized after no less than ten minutes had elapsed during which time the temperature of that fluid/component had not varied greater than ± 5°F. The test was terminated when all components of interest had stabilized or a critical component had exceeded its maximum specified temperature. The maximum temperatures permitted by the engine and transmission manufacturers for the applicable fluids are shown in Table 11.

Table 11
Maximum Permitted Temperatures
from Vendor Recommendations

Oil or Coolant Fluid	**Maximum Temperature**
Coolant fluid in radiator top tank	230°F
Coolant fluid out of radiator	<215°F
Engine Oil in Sump	275°F (300°F Absolute Maximum)
Torque Converter Out	275°F (300°F Absolute Maximum)
Oil Temperature Out to Heat Exchanger	300°F
Oil in Differentials	300°F
Fuel Temperature	150°F
Turbo Exhaust	1200°F (1300°F Absolute Maximum)
Oil in Power Steering	300°F

All tractive effort values were based on the designed CCGVW of 82,000 pounds. Using the tractive effort formula, *T.E.* x *GVW* = Net Drawbar, Table 12 was used to determine the tractive effort values to be run and the equivalent continuous grade.

Table 12
Tractive Effort Values

Tractive Effort (%)	**Pounds Resultant Net Drawbar**	**Equivalent Gradeability**
20	8,200	20.4
30	16,400	31.4
40	32,800	43.6
50	41,000	57.6
60	49,200	75.0

10.2 Full Load Cooling Test Procedures

10.2.1 Full Load Cooling Test Preparation

The LVSR-TD was re-weighed payloaded to the cross-country specification with 18 tons. Tables 4 and 5 show the vehicle weights.

All critical fluids were drained and refilled in accordance with component manufacturers' recommendations. A 50/50 ethylene glycol coolant mix was verified in the radiator. As stated in the preliminary lubrication order, a multi-viscosity synthetic oil (ExxonMobile Delvac-15W-40) was used to fill the single oil reservoir (engine, transmission, steering, fan, etc.).

Test instrumentation was installed on the vehicle. Table 13 lists the instrumentation installed.

With the exclusion of the turbo exhaust, all temperatures were measured with type "T" thermocouples (copper and copper-nickel). The turbo exhaust temperature was measured with a type "K" (nickel-chromium and nickel-aluminum) thermocouple. All thermocouples were calibrated against a reference prior to use, first in an ice bath and then in an oil bath through several points in their expected range of measurement. The coolant flow rate was recorded using a Sponsler flow transducer with an accuracy of ± 2%. RPM data was recorded from the DDEC IV software on the vehicle. Speed was recorded using an ADAT radar speed transducer. Drawbar load was measured with a Strainsert, 125,000 pound strain gauge load cell, with an accuracy of ± 1% of full scale, mounted in the drawbar cable between the test vehicle and the mobile dynamometer(s). All the above data were recorded on a MEGADAC digital data acquisition recorder. Full load cooling instrumentation occurred in August 2001 with the data channels listed in Table 13.

Table 13
Instrumentation Installed on LVSR-TD For Full Load Cooling Test

Channel	Location	In	Units
1	SPEED		MPH
J1939	ENGINE RPM		RPM
J1939	COOLANT FLOW		GPM
2	AMBIENT RADIATOR	Air	°F
3	CENTER OF RADIATOR ABOVE FAN	Air	°F
4	RIGHT RADIATOR	Air	°F
5	FRONT RADIATOR	Air	°F
6	LEFT RADIATOR	Air	°F
7	REAR RADIATOR	Air	°F
8	CHARGE AIR COOLER (CAC) INTO COOLER	Air	°F
9	CHARGE AIR COOLER (CAC) OUT FROM COOLER	Air	°F
10	RADIATOR COOLANT IN	Fluid	°F
11	RADIATOR COOLANT OUT	Fluid	°F
14	ENGINE OIL SUMP	Fluid	°F
15	TORQUE CONV OUT TO HEAT EXCHG	Fluid	°F
16	OIL OUT FROM HEAT EXCHG	Fluid	°F
17	AXLE#3 DIFFERENTIAL	Fluid	°F
20	TURBO EXHAUST	Air	°F
	DRAWBAR LOAD		LBS
	AMBIENT	Air	°F

A second readout of ambient temperature was recorded during the test using NATC's weather station data at the paved test facility.

10.2.2 Full Load Cooling Test Conduct

After the instrumentation process, all fluids were verified to be at their proper levels.

Full load cooling was to be performed in incremental steps up to 60% of vertical load. Using the specified design target CCGVW of 82,000 lbs., the specified drawbar value at 60% would be 49,200 lbs. Test operation was to be at wide-open throttle, in the lowest gear available. Depending upon the level of tractive effort desired, speed would be adjusted to reflect the required drawbar.

The LVSR-TD, payloaded to CCGVW payload, was connected to the mobile dynamometer(s) through a drawbar cable and appropriate capacity load cell. The vehicle was then operated in the lowest appropriate gear at wide-open throttle to achieve the desired drawbar value. The mobile dynamometer provided the resistance as measured through the tension load cell. The test continued in this mode until temperature stabilization was reached in each fluid or a particular component reached its critical temperature. A component was considered stabilized after no less than ten minutes had elapsed during which time the temperature of that component had not varied more than ± 5°F after correcting for ambient temperature change. The test was terminated when all components of interest stabilized or a critical component exceeded its maximum specified temperature. The tests were conducted at NATC's paved 1.8 mile oval test track.

10.2.3 Full Load Cooling Data Analysis Procedures

All fluid temperature data were processed either by correcting to the specified ambient temperature of 125°F., or, in the case of the coolant only, air-to-boil, which is the extrapolated temperature at which the coolant would boil. This latter method of treatment relates only to the coolant boil point and indicates how far the cooling system is from boiling under the condition of test. The boiling point of the coolant is derived based on the level of the pressure cap and the type of coolant. In the instance of the LVSR-TD, the coolant mixture was 50% ethylene glycol and 50% water. The pressure cap on the cooling system was rated at 9 pounds. The values used for coolant boiling points were derived from Figure 5-1 of AMCP 706-361, *Engineering Design Handbook, Military Vehicle Power Plant Cooling*, Headquarters, US Army Materiel Command, June 1975. With the nominal 9 pound pressure cap, the boiling point of a 50/50 coolant mix would be 251°F. Figure 1 below reproduces this chart.

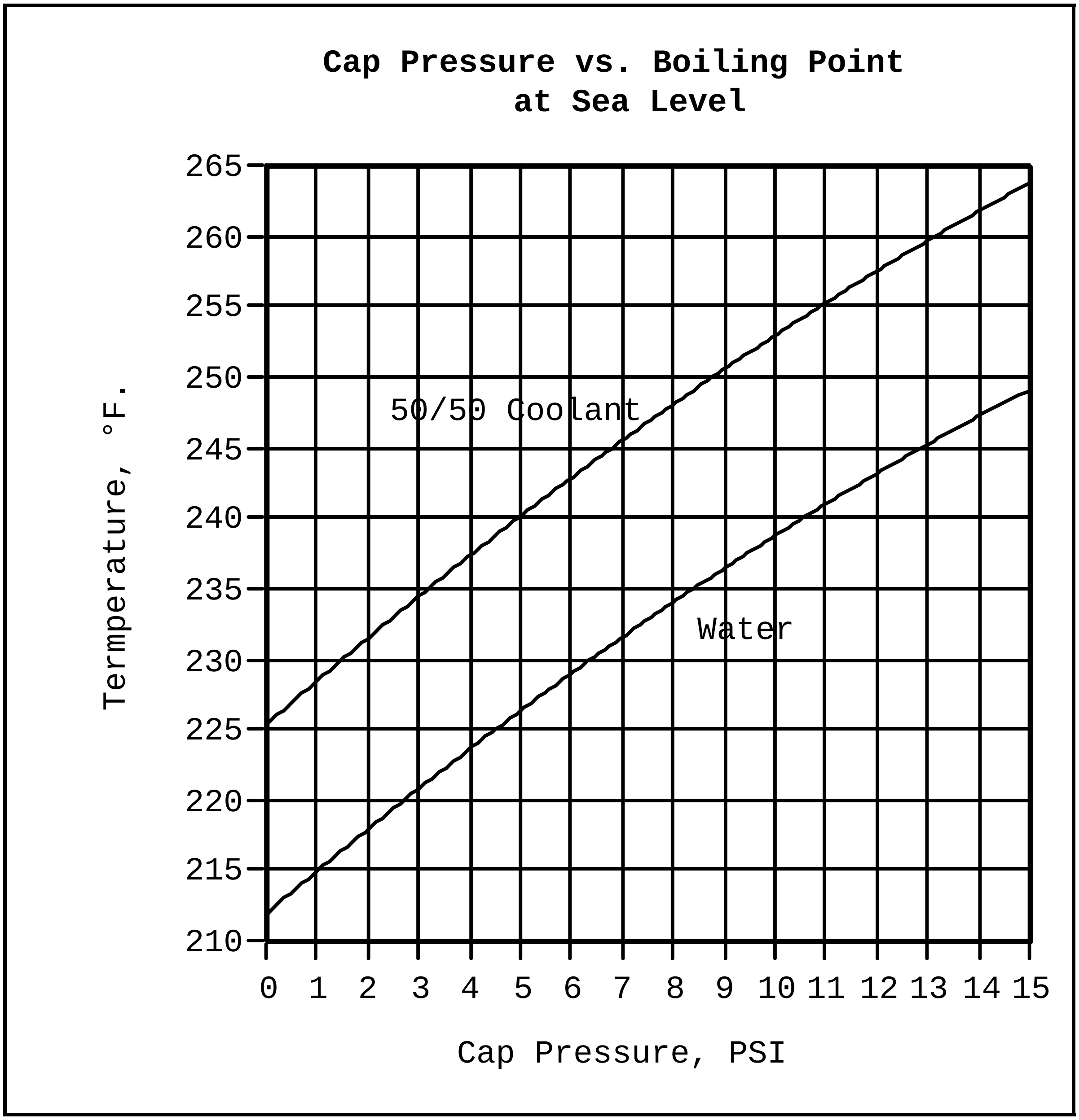


Figure 1
Cap Pressure vs. Boiling Temperature
(At Sea Level)

Coolant temperatures were also to be corrected for altitude. The source for the altitude correction was SAE J1393, *On-Highway Truck Cooling Test Code*, which permits a 2°F to 4°F correction in temperature per 1,000 feet in altitude when the test is conducted 500 feet above sea level. The test altitude for NATC is 4,200 feet above sea level. The maximum correction, or 16.8°F, can be used in all calculations. The rationale for this is that as altitude increases, air density decreases and reduces engine and cooling system performance. Therefore, these factors must be taken into consideration during analysis of coolant values achieved.

Coolant temperatures were corrected using the following formulae:

1. Corrected Temperature for Fluid Other Than Coolant:

   CT = FT + ST - AT

2. Corrected Coolant Temperature with Altitude Compensation:

   CT = FT - AC + ST - AT

3. Air-to-Boil with Altitude Compensation:

   ATB = AT + CB - (FT - AC)

Correction To A Specified Ambient Key:

- CT: Corrected Temperature
- FT: Recorded Fluid Temperature
- ST: Specified Ambient Temperature (125°F.)
- AT: Ambient Temperature
- AC: Altitude Correction (4°F per 1000 feet)

Air-to-Boil Key:

- ATB: Air-to-Boil
- FT: Recorded Coolant Temperature
- ST: Specified Ambient Temperature (125°F.)
- AT: Ambient Temperature
- AC: Altitude Correction (4°F per 1000 feet)
- CB: Coolant Boil Temperature

10.2.4 Full Load Cooling Test Results

Beginning on 30 August 2001 and continuing through 10 September 2001, a total of seven evaluations were performed to determine the minimum threshold at which the system would cool. The evaluations varied between 10% tractive effort incrementally up to 60% tractive effort. The 60% tractive effort test was repeated a second time to measure fuel temperature differences. The full load cooling data is provided in Appendix C.

Tables 14 to 17 summarize the temperatures, corrected to 125°F, which were recorded at the 10%, 30% and 60% tractive effort levels.

Table 14
Full Load Cooling Parameters at 10% Tractive Effort
Corrected to 125 Degree F Ambient
30 Aug 2001

Component	Maximum Avg Recorded Corrected Temperature (°F)	Maximum Specified Temperature (°F)
Engine Oil in Sump	301	300
Transmission Fluid from Torque Converter (Torque Converter Out)	260	300
Coolant to Radiator	238	230-240
Coolant Delta "T"	15	12-20
Oil Temperature Out to Heat Exchanger	257	300
Oil in Differentials	184	300
Turbo Exhaust	900	1300

Table 15
Full Load Cooling Parameters at 30% Tractive Effort
Corrected to 125 Degree F Ambient
31 Aug 2001

Component	Maximum Avg Recorded Corrected Temperature (°F)	Maximum Specified Temperature (°F)
Engine Oil in Sump	302	300
Transmission Fluid from Torque Converter (Torque Converter Out)	264	300
Coolant to Radiator	240	230-240
Coolant Delta "T"	17	12-20
Oil Temperature Out to Heat Exchanger	262	300
Oil in Differentials	190	300
Turbo Exhaust	950	1300

Table 16
Full Load Cooling Results at 60% Tractive Effort Corrected to 125 Degree F Ambient
07 September 2001

Component	Maximum Avg Recorded Corrected Temperature (°F)	Maximum Specified Temperature (°F)
Engine Oil in Sump	308	300
Transmission Fluid from Torque Converter (Torque Converter Out)	279	300
Coolant to Radiator	250	230-240
Coolant Delta "T"	19	12-20
Oil Temperature Out to Heat Exchanger	272	300
Oil in Differentials	205	300
Turbo Exhaust	950	1300

10.2.4.1 Fuel Temperature Modification Test

The maximum specified fuel temperature to the engine was defined to be 140 degrees F. This is to give both the horsepower expected for the volume of fuel as well as protect the fuel system components. During the 0.1, 0.3 and 0.6 tractive effort tests, the fuel on the DDEC was 150 degrees F (flat panel readout) and the fuel tank was 120 degrees F based on a handheld thermocouple reading after these runs. The flashpoint is approximately 176 degrees F for diesel and 170 degrees F for JP-8.

To reduce the recorded fuel temperatures, NATC investigated a fuel cooler but found this to be an unacceptable approach. In lieu of a fuel cooler, NATC installed a turbo blanket to minimize turbo charger radiant heat. NATC also improved the shielding at the passenger side fuel tank, the tank closest to the turbocharger. The 0.6 tractive effort test was run a second time. The fuel temperatures dropped approximately 10 degrees to an acceptable range. Table 17 shows the corresponding temperatures at the other critical locations.

Table 17
Full Load Cooling Results at Second 60% Tractive Effort
Corrected to 125 Degree F Ambient
10 September 2001
(Turbo Blanket Installed and Fuel Tank Shielding)

Component	Maximum Avg Recorded Corrected Temperature (°F)	Maximum Specified Temperature (°F)
Engine Oil in Sump	306	300
Transmission Fluid from Torque Converter (Torque Converter Out)	266	300
Coolant to Radiator	240	230-240
Coolant Delta "T"	18	12-20
Oil Temperature Out to Heat Exchanger	265	300
Oil in differentials	180	300
Turbo Exhaust	1000	1300

10.3 Full Load Cooling Conclusions

The coolant temperatures into the radiator showed marginal performance after it was corrected for ambient and altitude. However, the safety features built into the electronically controlled engine did not allowed the actual temperature to rise above manufacturer recommended temperatures. It is probable that this technology would prevent the coolant temperature from rising above the maximum specified temperature if the test were performed at an actual ambient temperature of 125 degrees F. This is indicated by the dip in the temperatures in the graphs in Appendix C where the engine went into a de-rate mode.

In concurrence with this, the engine oil sump temperatures were shown to be marginal after the ambient adjustment, either at or slightly below the maximum temperature specified (300 degrees F). This temperature might also be regulated with the engine’s de-rate function at actual ambient temperatures of 125 degrees F. All other fluids monitored during the tractive effort test were found to be well within their respective limits.

The only other data shown to be outside the parameters set forth by the manufacturer was the engine coolant delta. The recommended manufacturer delta was 12 degrees F. The LVSR-TD had a coolant delta of 19 degrees F. However, this was still within an acceptable range of 12 to 20 degrees. The photographs for the Full Load Cooling test are contained in Appendix B, Photograph Numbers 20238-014 through 20238-019.

11.0 RIDE QUALITY

The ride quality evaluation was performed to determine if the vehicle was in compliance with the "Ride Quality" paragraph of the draft Performance Specification. This paragraph states that "all vibration and acceleration measurements shall be taken on the cab floor at the driver's station, on the driver's seat, and in the center of the cargo bed. All variants shall attain no more than 6 watts average vertical absorbed power at the driver's station and at the driver's seat with the seat locked out while negotiating a course with the road roughness values and at speeds per Table 3-3."

Performance Specification Table 3-3-- Ride Quality

RMS (in)	@ GVW	@ CW
	MPH	MPH
<0.3	65	65
0.4	60	60
0.6	25	25
1.5	20	20
2.0	15	15
3.0	5	5

The ride quality evaluation was also performed to determine if the vehicle was in compliance with the "Shock" paragraph of the draft Performance Specification. This paragraph states that "all variants shall attain no more than 2.5 g vertical acceleration on the cab floor at the driver's station, on the driver's seat, and in the center of the cargo bed while negotiating half-round obstacles of 10 inch heights at speeds up to 20 MPH and 12 inch heights at speeds up to 10 MPH with tires at normal cross-country inflation pressure.

11.1 Ride Quality

The courses associated with ride quality tests are designed to identify the dominant frequencies and accelerations in those parts of a vehicle that directly or indirectly affect the ride quality for human occupants and/or cargo. Ride quality measurements can be used to isolate problem areas of suspension performance by identifying parts of a vehicle that contribute to ride harshness over rough terrain. One such measurement is absorbed power. Absorbed power is a measure of acceleration in three orthogonal axes at the driver’s station and is a function of the acceleration input and its frequency content. This method requires the calculation of the power (in watts) absorbed by the individual seated in the driver's station and at the base of the seat. The absorbed power is calculated by multiplying the Power Spectral Density (PSD) by certain weighting factors. These factors are different for each of the three mutually perpendicular axes and give the highest weighting to frequencies that do the most damage to the human body from a ride quality perspective.

The photographs for the Ride Quality and Handling test are contained in Appendix B photographs 20238-020 through 20238-023.

11.1.1 Ride Quality Instrumentation

Prior to conducting the dynamic ride quality engineering test, the vehicle was instrumented with the channels of data listed in Tables 18 and 19.

Table 18
LVS Ride Quality/Model Validation Instrumentation

Channel	Location and Orientation	Units
1	Speed	MPH
2	#1 axle right side vertical accelerometer	g
3	#3 axle right side vertical accelerometer	g
4	#4 axle right side vertical accelerometer	g
5	Front frame vertical accelerometer	g
6	Rear frame vertical accelerometer	g
7	Passenger side seat base vertical accelerometer	g
8	Passenger side seat cushion vertical accelerometer	g
9	C.G. pitch rate	deg/s
10	#1 axle to frame vertical displacement	inch
11	#2 axle to frame vertical displacement	inch
12	#3 axle to frame vertical displacement	inch
13	#4 axle to frame vertical displacement	inch
14	RBU Left front frame rail strain	µS
15	RBU Right front frame rail strain	µS
16	RBU Left rear frame rail strain	µS
17	RBU Right rear frame rail strain	µS

Table 19

LVSR-TD Variants Ride Quality/Model Validation Instrumentation
Instrumentation Also Transferred to Raydan and Hendrickson RBUs

Chan.	Location and Orientation	Units
1	Driver seat base vertical accelerometer	g
2	Driver seat base lateral accelerometer	g
3	Driver seat base longitudinal accelerometer	g
4	#1 axle left side vertical accelerometer	g
5	#3 axle left side vertical accelerometer	g
6	#5 axle left side vertical accelerometer	g
7	Frame at #1&2 center vertical accelerometer	g
8	Frame at #4 axle left side vertical accelerometer	g
15	#1 axle left side wheel travel	deg
16	#3 axle left side wheel travel	deg
17	#5 axle left side wheel travel	deg
22	Speed	mph

11.1.2 Ride Quality Test Procedures

The ride quality tests had two objectives, to measure the absorbed power at the driver station and to validate the ADAMS models of the various LVSR variants.

For this test, the vehicles were run over courses that are representative of the LVSR mission profile and over standard RMS ride quality courses. These courses vary in terms of road roughness amplitude and frequency. The vehicles were run over the courses at different speeds. If a vehicle resonance frequency was identified, the vehicle was driven at speeds below and above the resonance frequency. This allows for an analysis of absorbed power at a range of speeds recognizing that the absorbed power peaks at the resonance frequency condition.

For the LVSR-TD model validation, time histories, PSD plots and transfer functions were generated for each accelerometer channel. PSDs allow the comparison of amplitude versus frequency. This analysis allowed analysis of the steady state and resonance frequency ride quality conditions. Transfer functions describe the gain or attenuation of acceleration energy from one accelerometer location to another. If the gain is greater than one, the output is greater than the input. The phase angle between locations was also computed. If the phase between locations is 0 degrees or 360 degrees, the components are accelerating in the same direction (in phase). If the phase angle is 180 degrees, the components are accelerating in opposite directions (out of phase). The transfer function relationships listed in Table 20 were computed and analyzed for model validation.

Table 20
Transfer Function Relationships

Input Acceleration	**Output Acceleration**
#1 Axle	Front Frame
#1 Axle	Seat Base
#3 Axle	Rear Frame
#5 Axle	Rear Frame (LVSR-TD only)
Front Frame	Seat Base
Front Frame	Seat Cushion
Seat Base	Seat Cushion

For the LVSR draft Performance Specification validations, absorbed power was the ride quality measurement of interest. The vertical acceleration measured at the seat base was processed and filtered with respect to human response vibration conditions as outlined in the Army absorbed power program or ISO 2631-1: 1997, then summed and averaged over the entire length of the course.

11.1.3 Ride Quality Analysis

Ride quality measurements are typically measured and presented as relationships of speed (mph) versus terrain roughness (inches of RMS). The ride quality curves are based on six watts of absorbed power at the driver's station. Six watts represents an energy input to the driver that can be sustained for an extended period without decreased driving proficiency due to fatigue or injury. Additionally, as the RMS value increases, the physical road roughness increases. Pavement is generally between 0.1 and 0.5 inch RMS. In the paved region of the curve, the vehicle is generally limited by powertrain performance or maximum speed of the vehicle. Gravel roads are generally between 0.3 and 1.0 inch RMS. Trails are between 1.0 and 3.4 inches RMS and cross-country is between 1.5 and 4.8 inches RMS. Any terrain roughness over approximately 1.0 inch starts to limit vehicle speed based on the vehicle’s ability to negotiate the roughness in the road while still maintaining the six watt ride quality.

The measurement of absorbed power (watts) is accomplished with a vertical accelerometer mounted at the base of the driver’s seat. Processing the vertical acceleration in conjunction with the weighting filters for the sensitivity of the human body yields the absorbed power in watts. Various test courses with measured

RMS roughness levels were run at increasing speeds to find ride levels below and above the six watt level. Plotting a curve for ride quality (in watts) versus speed (in MPH) allows the definition of the speed that will produce the six watt level. The speed for the first occurrence of six watts was selected. For several of the ride quality courses, the six watt criteria was limited by a resonance condition at an intermediate speed and the absorbed power actually decreased with increased speed.

11.1.3.1 LVSR Configurations

Three different LVSR RBUs were evaluated for ride quality. The LVSR-TD vehicle with the independent suspension at all axle locations was evaluated. Second, the RBU built with a Raydan air ride suspension was installed on the LVSR-TD FPU and tested over the same ride quality courses. Finally, the RBU built with a Hendrickson HHP hydraulic suspension was installed on the LVSR-TD FPU and tested over the same ride quality courses. This allowed a comparison of ride quality based on a passive independent suspension, a constant ride height air ride suspension and a constant ride height hydraulic suspension at two payload configurations. Each configuration was a 10x10 vehicle configuration. The 22 ton payload (HGVW) was not run as the 18 ton payload (CCGVW) is the maximum off-road payload.

The LVSR-TD was a passive independent suspension with fixed rate spring elements. Passive independent suspensions ride higher unloaded than loaded. Suspension design requires spring elements designed for the heaviest load at ride height. The LVSR-TD was designed to an 18 ton design load (16 1/2 ton payload plus 1-1/2 ton flatrack) at a 12 inch ride height. Optimizing tire pressure, at low payload compensates for problems associated with empty vehicle ride, but as the weight differential increases, tire pressure alone will not offset unloaded suspension behavior.

The LVSR program also demonstrated two suspension systems that maintain a constant ride height. Again, these suspensions included the Raydan air ride suspension and the Hendrickson HHP hydro-pneumatic strut suspension. Two different RBUs were built with these suspensions and the original LVS solid axles. A third axle was added to the RBU of each variant to compensate for the 18 ton payload requirement. Air and hydro-pneumatic spring elements permitted controlling ride height based on payload changes.

11.1.3.2 Ride Quality Evaluation Results – Six Watt Criteria

Table 21 and Figures 2 through 6 show the average speeds for the six watt average vertical absorbed power at the driver’s station (base of seat) with the tires at the appropriate tire inflation pressures. This was conducted at the CCGVW and empty vehicle configurations for road roughness values ranging from 0.5 to 4.4 inches. For initial comparisons (Figures 2 and 3), the LVSR-TD results are shown against the MTVR requirements.

Table 21
Loaded (CCGVW) and Empty Ride Quality for LVSR-TD (Independent), Raydan (Air) and Hendrickson (Hydraulic) Rear Body Units

Variant:	LVSR-TD	LVSR-TD	Raydan	Raydan	Hendrickson	Hendrickson
FPU:	Independent	Independent	Independent	Independent	Independent	Independent
RBU:	Independent	Independent	Air	Air	Hydraulic	Hydraulic
Payload:	18T	Curb	18T	Curb	18T	Curb
RMS Roughness	Speed	Speed	Speed	Speed	Speed	Speed
(Inches)	(MPH)	(MPH)	(MPH)	(MPH)	(MPH)	(MPH)
0.5	26	16	30	32.5	30	21
1	21	14	23.5	18.5	23.5	17.5
1.65	16	12	14.75	15	17.5	16.5
2.4	11	11	10.5	11	15	11.5
3.6	12.7	13	12.25	10.75	13.5	10.5
4.4	12.1	12.5	11.25	10	12.5	9.5

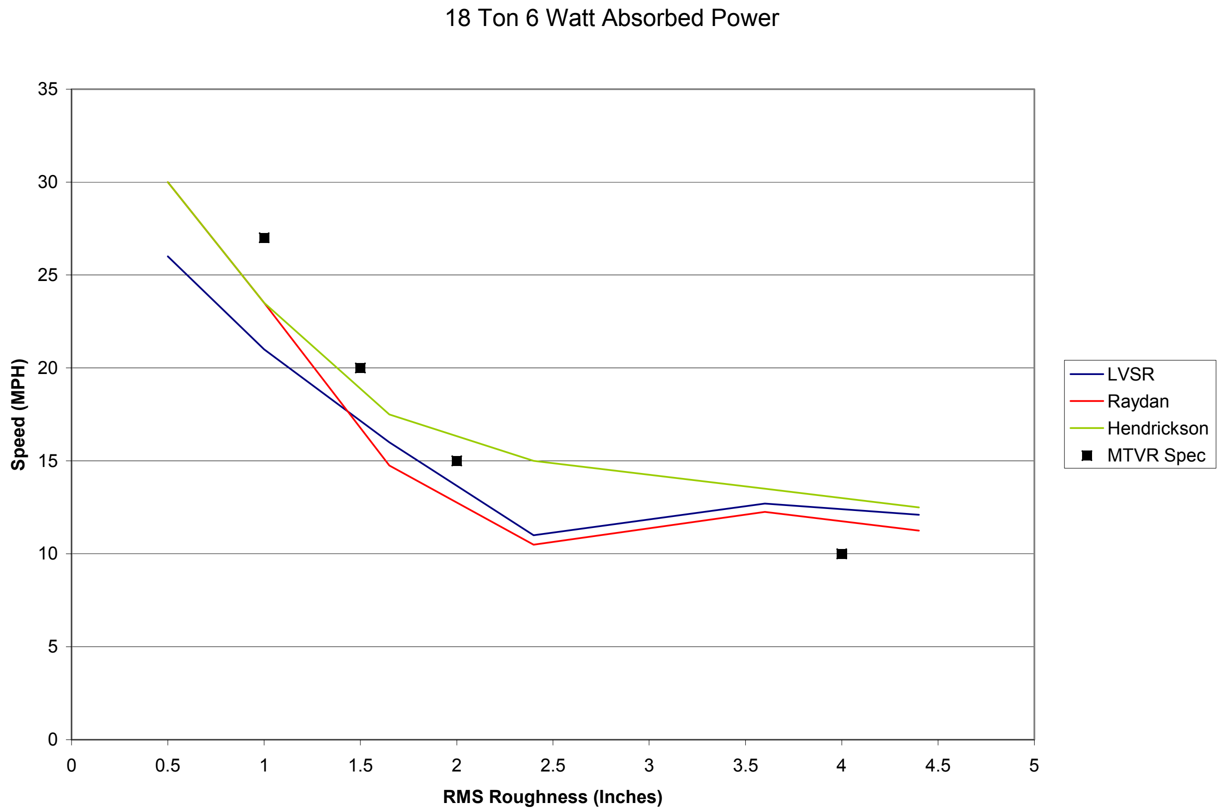


Figure 2
CCGVW Ride Quality for LVSR-TD (Independent), Raydan (Air) and Hendrickson (Hydraulic) Rear Body Units for 6 Watts of Absorbed Power at Base of Driver's Seat

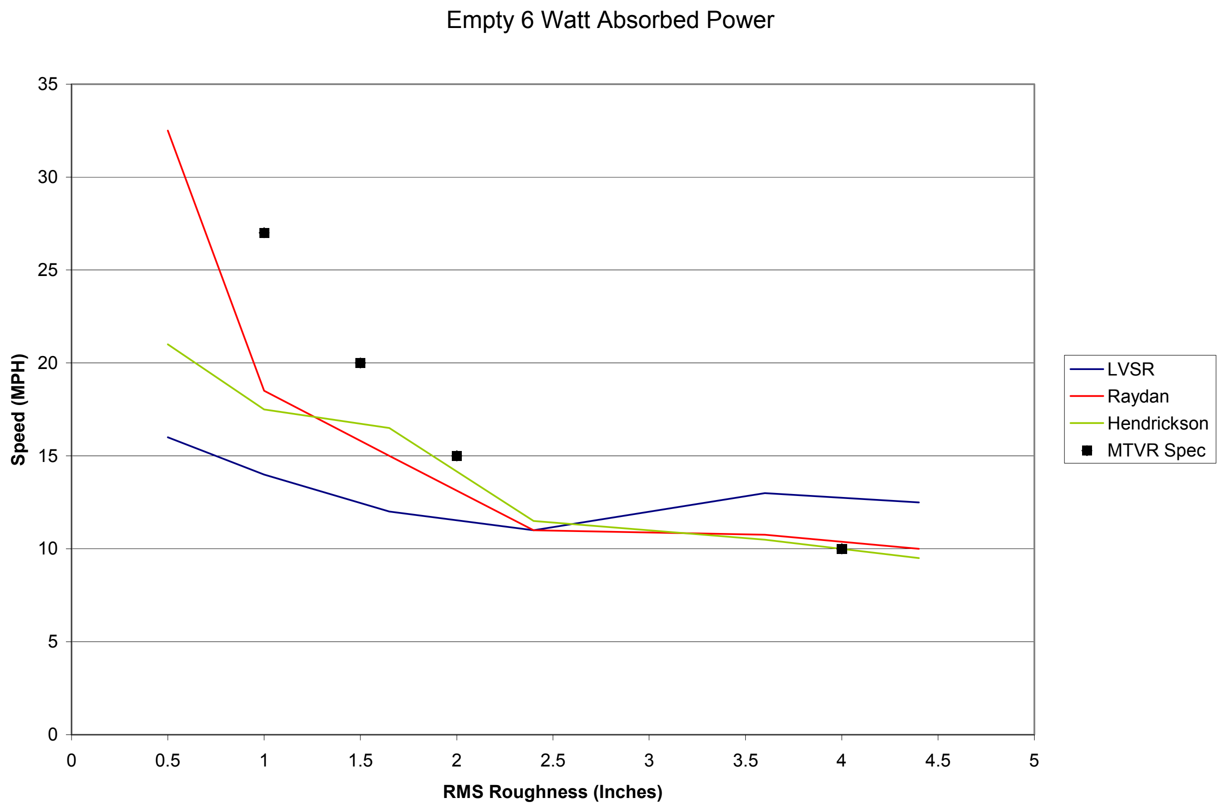


Figure 3
Empty Ride Quality for LVSR-TD (Independent), Raydan (Air) and Hendrickson (Hydraulic) Rear Body Units for 6 Watts of Absorbed Power at Base of Driver's Seat

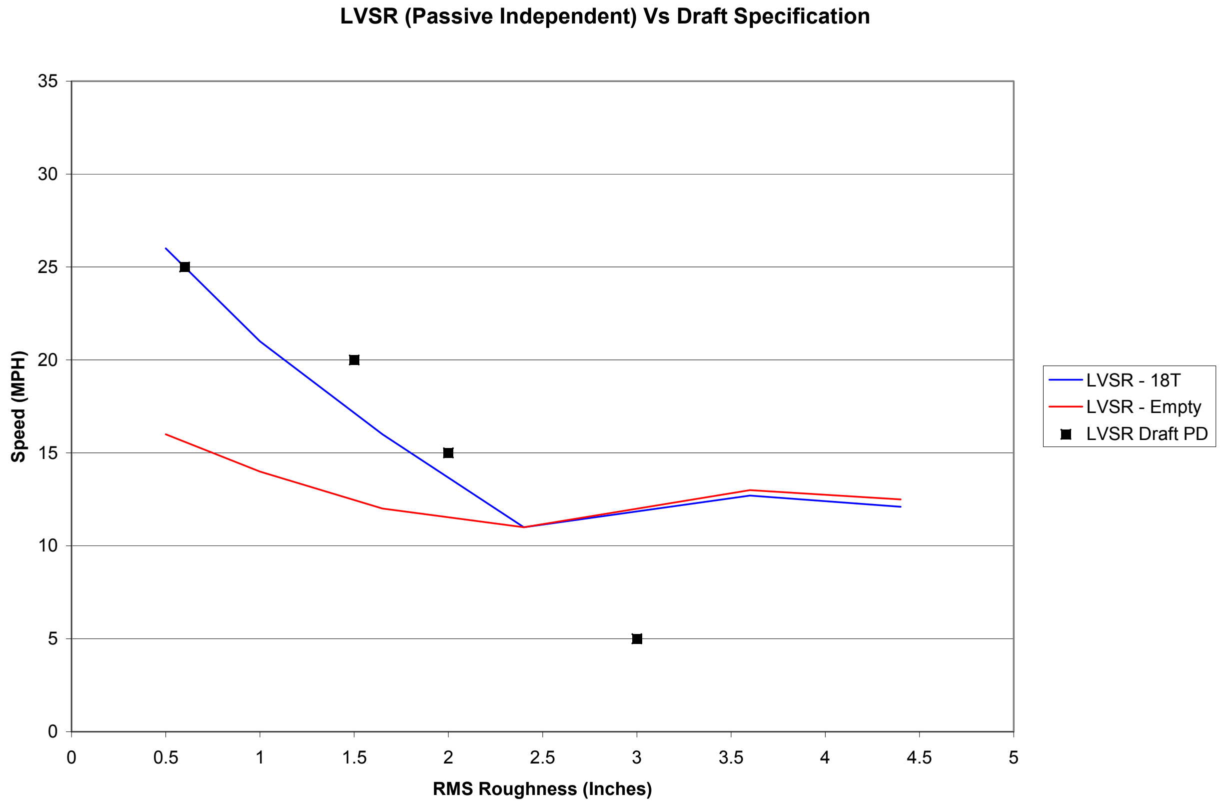


Figure 4
Ride Quality Comparison for LVSR-TD (Passive Independent) Versus LVSR Draft Performance Specification for 6 Watts of Absorbed Power at Base of Driver's Seat

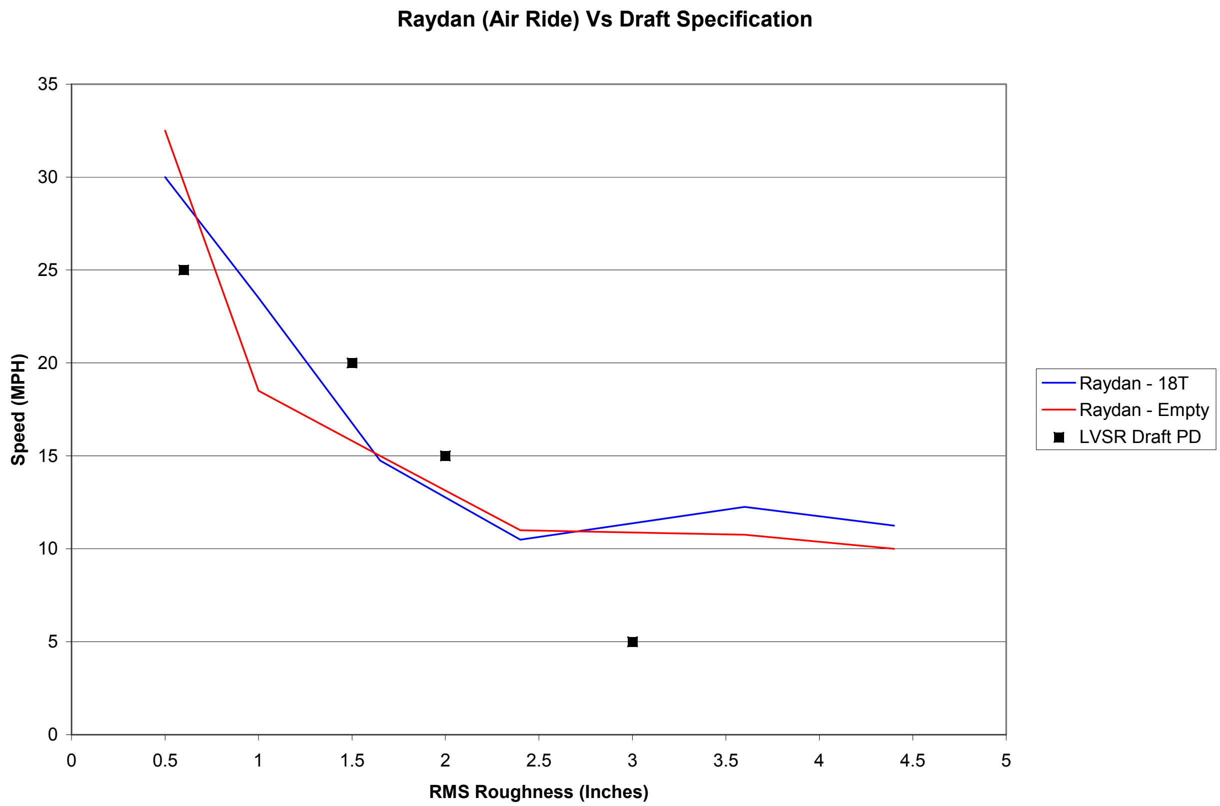


Figure 5
Ride Quality Comparison for Raydan Air Ride on RBU Versus LVSR Draft Performance Specification for 6 Watts of Absorbed Power at Base of Driver's Seat

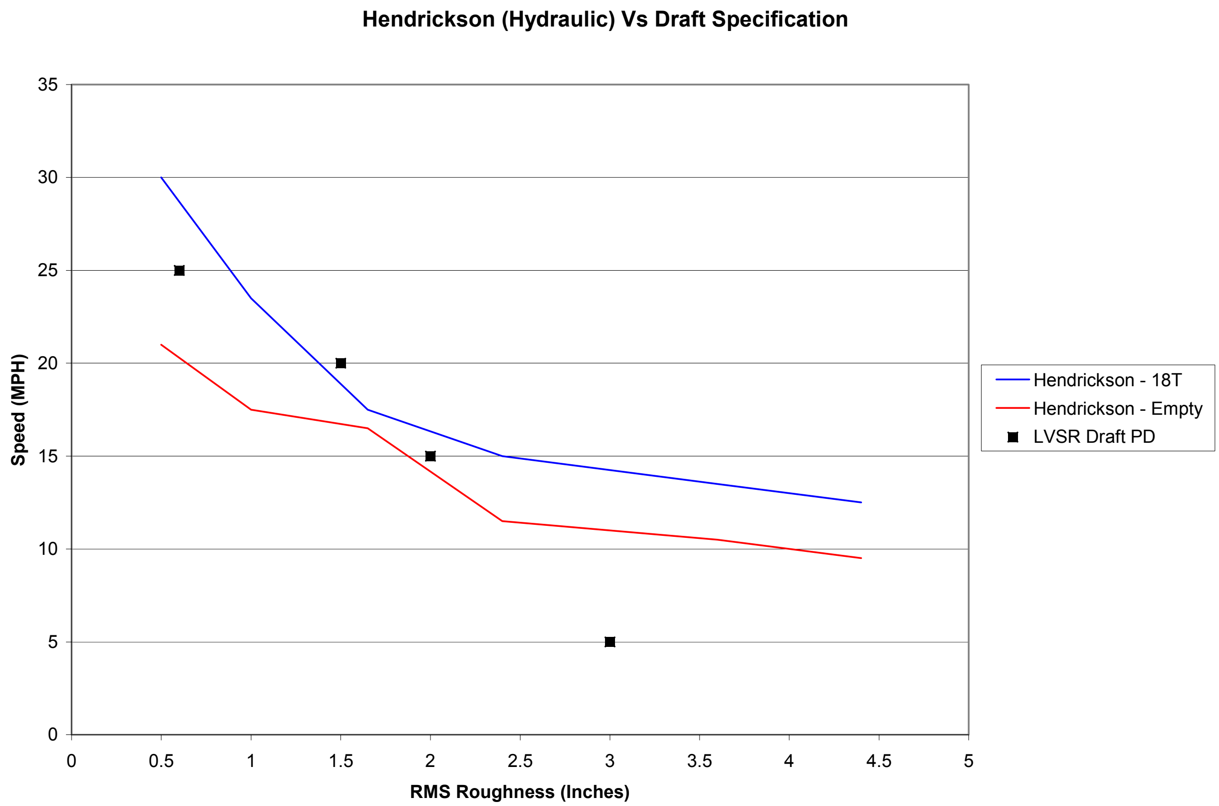


Figure 6
Ride Quality Comparison for Hendrickson Hydraulic Strut on RBU Versus LVSR Draft Performance Specification for 6 Watts of Absorbed Power at Base of Driver's Seat

Note to Figures 4 - 7: The LVSR must be able to operate at 10 MPH on 3 inch RMS without required six watt ride quality.

Each of the three suspension configurations were evaluated "as built" and no suspension tuning or suspension optimization was performed before the ride quality evaluation was conducted. For example, the LVSR-TD suspension was nearly identical to the MTVR suspension and had the MTVR springs installed. The front shock absorbers were MTVR front shocks. In the rear suspension of the LVSR-TD, Qatar independent suspension HEMTT shocks are used. It was known that suspension and shock absorber tuning would lead to changes in spring and shock absorber rates for optimized suspension performance.

Using the computer models validated with the above instrumentation, the LVSR-TD was optimized in the model and run over the same ride quality courses and half-round events. The suspension was optimized to produce the best combination of empty and loaded ride quality, empty and loaded shock attenuation (10 inch half-round) and stability events. No modifications were made to cab configuration or cab mounting within the modeling and simulation environment to improve the driver's station ride quality. Figure 7 shows the six watt absorbed power ride quality curve representative of improvements in the suspension tuning for the empty and loaded.

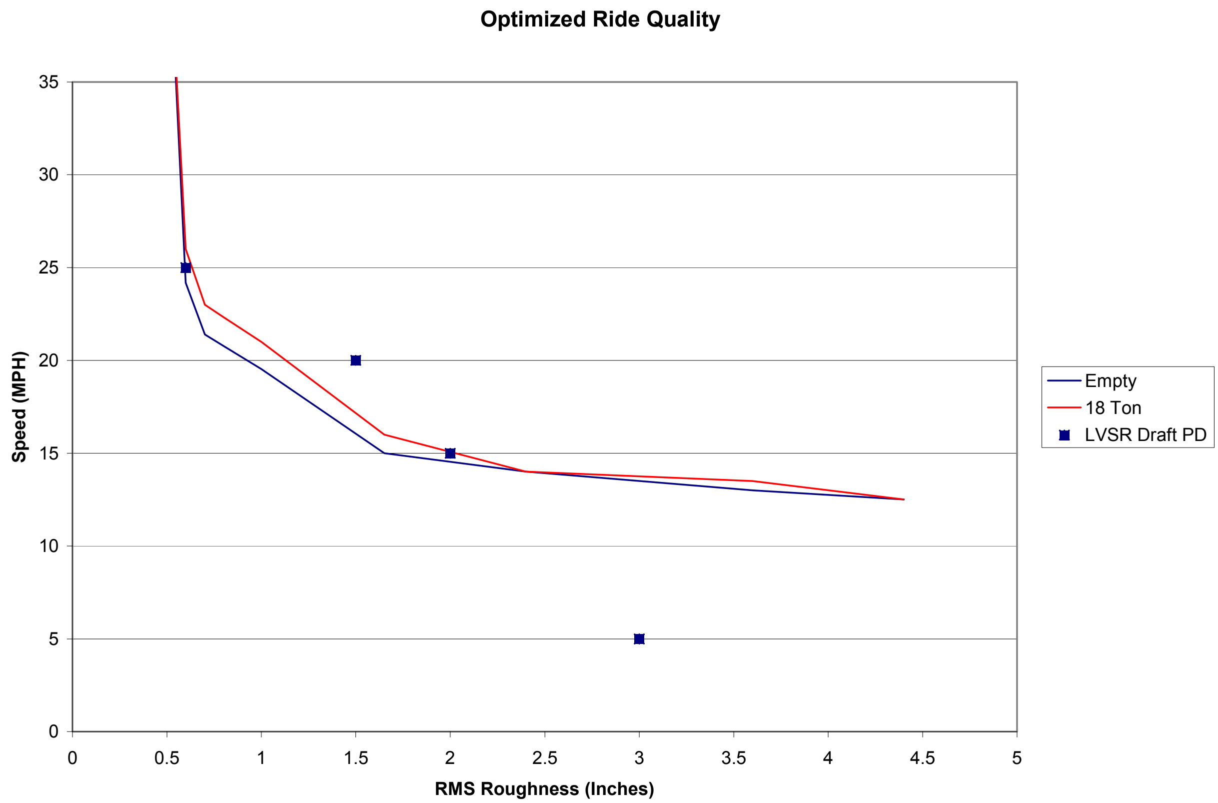


Figure 7
Optimized Ride Quality Comparison for LVSR-TD Versus LVSR Draft Criteria Performance Specification for 6 Watts of Absorbed Power at Base of Driver's Seat

11.1.3.3 Half-Round Evaluation

Another ride quality metric evaluated was the ability of the vehicle to attenuate shock inputs to the suspension and driver's station. The negotiation of half-round events at increasing speeds was run to find the limiting speeds based on a 2.5 g shock limit. The half-round requirement is that no more than 2.5 g of vertical acceleration is attained in either the positive or negative direction at the base of the driver's while negotiating half-round obstacles of 6, 8, 10 and 12 inches high.

Three different LVSR Rear Body Units (RBU) were evaluated for half-round inputs of 6, 8, 10 and 12 inches high. The LVSR-TD vehicle with the independent suspension at all axle locations was evaluated. The RBU built with a Raydan air ride suspension was installed on the LVSR-TD FPU and an RBU built with a Hendrickson HHP hydraulic suspension was installed on the LVSR-TD FPU. This allowed a comparison of ride quality based on an empty and 18 ton payload vehicle with constant ride height suspension alternatives. The results of the half-round test is shown in Figure 22.

Table 22
Speeds for 2.5 g at the Driver's Station
(Required 10 inch Half-Round at 20 MPH and
12 inch Half-Round at 10 MPH)

	LVSR Vehicle Variant					
Half Round Height	LVSR - 18T	LVSR - E	Raydan - 18T	Raydan - E	Hendrickson - 18T	Hendrickson - E
(Inches)	Speed (MPH)	Speed (MPH)	Speed (MPH)	Speed (MPH)	Speed (MPH)	Speed (MPH)
6	26.2	17.4	30.3	14.9	22.5	22.3
8	17.4	19.4	7.1	6.9	19.3	24.2
10	16.0	13.8	5.8	4.6	16.6	9.5
12	14.6	13.1	4.6	4.1	11.1	5.9

Similar to the ride quality analysis, the LVSR-TD suspension was optimized in the model and run over the same half-round events. The suspension was optimized to produce the best combination of empty and loaded ride quality, empty and loaded shock attenuation (10 inch half-round) and stability events. No modifications were made to cab configuration or cab mounting within the modeling and simulation environment to improve the driver's station ride quality. With improvements in the suspension tuning for the empty and loaded operation, the LVSR-TD was able to meet the draft Performance Specification requirement of 20 MPH for the 10 inch half-round height with the tires at CCGVW tire pressures. The Hendrickson RBU variant, similarly optimized within the modeling and simulation environment was able to meet the 10 inch half-round requirement.

11.1.4 Ride Quality Conclusions

Given a combination of physical testing and modeling and simulation with a validated model, the LVSR-TD met the draft

Performance Specification requirements for the six watt absorbed power and 2.5 g ride quality events. Additional mount tuning would improve the response on the 1.5 inch RMS roughness level for the 6 watt ride criteria.

12.0 FUEL ECONOMY TESTING

The fuel economy evaluation was performed to determine if the vehicle was in compliance with the "Fuel Economy" paragraph of the draft Performance Specification which states that the LVSR cargo variant shall achieve a minimum of 2.5 miles per gallon (MPG) over mission profile representative terrain.

The fuel economy test was conducted from 26 February through 6 March 2001. The mission profile for the LVSR-TD was segmented into the four terrain categories (paved, gravel, trails and cross-country) and the three payloads (empty, 18 tons and 22 tons). The combination of the 22 ton payload on trails and cross-country is not required for the LVSR-TD vehicle thus this combination was not run. This data and the individual fuel economy results are shown in Table 23. The following mission profile courses were run.

Surface Type	CTIS Setting	Speed Range
Paved	Highway	45-65 mph
Paved Test Track		
Gravel	Cross-Country	25-35 mph
Gravel Oval		
Perryman I		
Up and Down 15% Grade		
Trail/Cross-Country	Multi-Terrain	15-25 mph
Perryman III		
Alternating Bumps		
North Butte		
Forest Service Loop		
Sand Serpentine		
Susan's Bluff Loop (natural grades to 37%)		

Table 23
Fuel Economy Data and Results

Terrain	Fuel (Gallons)	Distance (Miles)	MPG	Percent of Mission
Highway				
22 Ton	10	46.3	4.6	18
18.5 Ton	19.9	46.1	2.3	2
Empty	15.6	44.3	2.8	20
Gravel				
22 Ton	23	41.4	1.8	10
18.5 Ton	20	41.5	2.1	10
Empty	19.1	43	2.3	10
Cross Country				
22 Ton	N/A	N/A	N/A	N/A
18.5 Ton	75.3	124.7	1.7	15
Empty	20.7	37.4	1.8	15

Average MPG Highway	3.0
Average MPG Gravel	2.0
Average MPG Cross-Country	1.7
Average MPG for Mission Scenario	2.6

The fuel economy value of 4.6 MPG for the highway portion at 22 tons is efficient because the engine is operating at its peak efficiency and the tire pressures are at their maximum inflation pressure (Table 23). When combined in the LVSR mission profile percentages, the average fuel economy was 2.6 MPH (Table 23 and Figure 8).

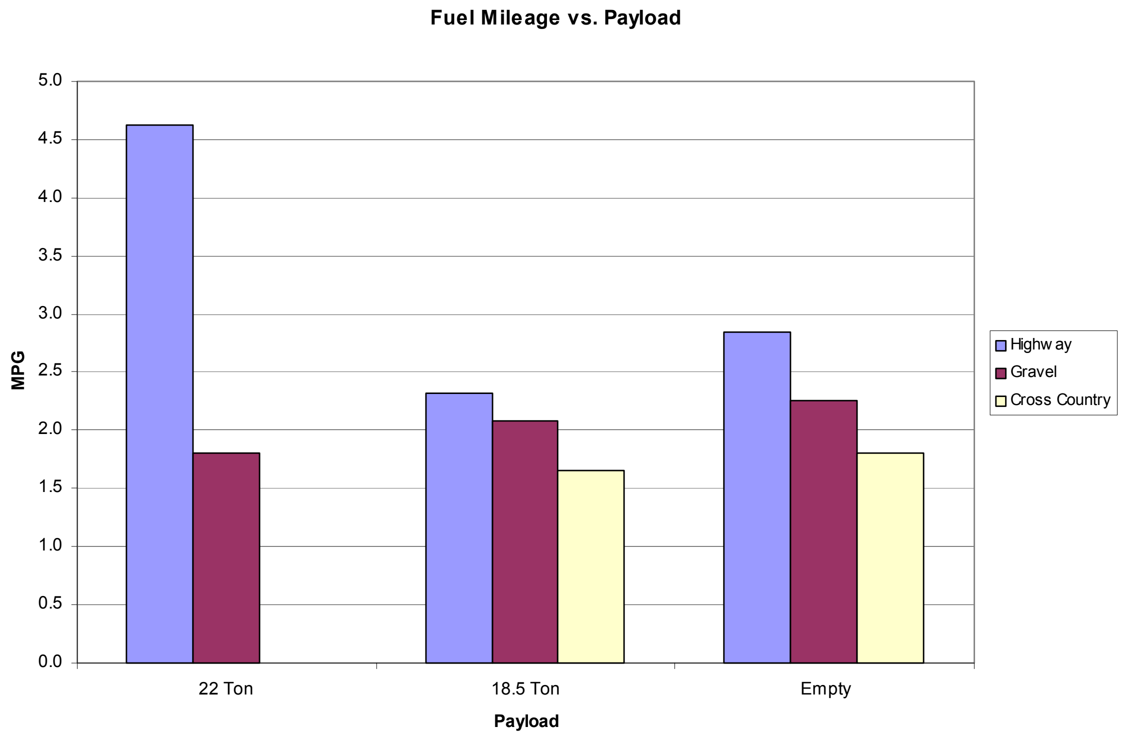


Figure 8
Fuel Economy Results By Payload and Mission Profile Terrain Type

Additionally, during the EOA held for the operator's evaluation and maintainer's evaluation between July 24 and August 11, 2000, the fuel economy of the LVS was calculated. The vehicle was operated over USMC mission scenarios for this period of the evaluations. The LVS achieved 2.1 MPG. The vehicle was operated empty and at a 12.5 ton payload.

12.1 Fuel Economy Conclusion

With a 40/30/20/10 terrain mix, the LVSR-TD met the draft Performance Specification fuel economy requirement of 2.5 MPG with diesel fuel.

13.0 PERFORMANCE TESTING

A range of performance testing was conducted to support an AoA for five LVSR alternatives and the required MOP for each. The performance testing was also performed to determine if the vehicle was in compliance with the various performance paragraphs of the draft Performance Specification. Finally, the

EOA operations were used to confirm many of the mobility performance requirements within the LVSR mission profile.

This performance testing was completed in late 2000 and early 2001 following the USMC EOA operations. As such, data for grade operation, side slope, speed, speed on grade, acceleration, range, ride quality, shock and vibration, steering and handling, turning radius and dimensions for transportability were accomplished to support the MOP data for the AoA. Additionally, the LVSR-TD involved an extensive J1939 electrical integration with demonstration of diagnostics and prognostics during the EOA and AoA.

Appendix G is a summary of the AoA MOP data. For correlation to this test report, the “Rebuy New Truck” alternative is the LVSR-TD. The “Remanufacturer” alternative was a combination of the test experience and results with the Raydan and Hendrickson RBUs.

As stated in Section 6.0, all performance tests were run with DF-2 diesel. A comparison acceleration test was performed to provide baseline data on the acceleration differences of the LVSR-TD with both DF-2 diesel and JP-8 fuel. The acceleration comparison test was conducted on the LVSR-TD in May 2001 at NATC’s paved 1.8 mile test track. The LVSR-TD was payloaded to its CCGVW and the tire pressures were set to the paved CTIS setting. The results are shown in Table 24.

Table 24
LVSR-TD Acceleration Diesel vs JP-8

Time Increment (MPH)	Run 1 Time (sec)	Run 2 Time (sec)	Run 3 Time (sec)	Run 4 Time (sec)	Average Time (sec)
Diesel (DF-2)					
0-15	5.6	5.7	6.1	5.6	5.8
15-25	6.4	6.1	6.1	6.0	6.2
25-45	30.1	30.2	29.0	28.5	29.4
45-50	-	13.9	-	13.0	13.5
0-45	42.1	42.0	41.2	40.0	41.4
JP-8					
0-15	5.9	6.2	6.0	6.9	6.3
15-25	6.3	6.6	6.1	7.5	6.6
25-45	28.0	28.9	30.5	33.1	30.1
45-50	12.2	12.1	14.0	14.0	13.1
0-45	40.2	41.7	42.6	47.5	43.0

13.1 Performance Testing Conclusion

The data showed the use of JP-8 over diesel degraded the acceleration performance approximately 4 percent. The LVSR-TD met the other performance requirements in the draft Performance Specification as shown in data results in Appendix G, using diesel (DF-2) fuel.

14.0 STABILITY AND HANDLING

The purpose of this evaluation was to provide baseline data on the static stability and controllability characteristics of the LVSR-TD at curb weight and payloaded configurations against the "Steering and Handling" requirements of the draft Performance Specification.

The tilt evaluation was performed on the vehicle with both the driver and passenger sides of the vehicle in the downslope direction. Following the static stability, dynamic stability testing included 200 foot constant radius curve, double lane change, side slope performance and stability and control braking in a wet 500 foot radius curve.

14.1 Tilt Test Methodology

The tilt table testing was conducted utilizing both NATC and SAE procedures. Because the tilt table facility utilized is an outdoor facility, the wind velocity was monitored throughout the tilt table test. The tilt table test was conducted only when the wind velocity was less than 10 MPH.

In accordance with the test requirements, the LVSR-TD was positioned on the tilt table in a straight line parallel to the tilt axis. This test was conducted for only the LVSR-TD vehicle.

During tilting, the vehicle was restrained to prevent it from rolling off the tilt table. The restraints were set such that the axles could lift approximately six inches off of the table surface. The tilt table is covered with a high friction surface to prevent the vehicle from sliding down the table during tilting (Reference Photograph Numbers 20238-24 and -25).

14.1.1 Tilt Table Test Conduct

The tilt evaluation was performed on the LVSR-TD with both the driver and passenger sides of the vehicle in the downslope direction. The vehicle was tilted three times in each direction for each configuration except for the payloaded, cross-country tire pressure, passenger side down configuration.

During each tilt, inclinometers were mounted on the front bumper, axle #1, rear bumper, and axle #5 in order to measure the angles at these locations. One inclinometer secured to the tilt table was used to measure table angle. A digital bubble level secured to the table was used to monitor table angle during the test. The inclinometer data was processed as angle versus time for each tilt and the points of initial lift of each wheel are identified. The average lift angle for the tilts in each direction was used to calculate the "rollover threshold," which is equal to the tangent of the table angle at the point of instability of the vehicle. For the smaller angles generally experienced by heavy vehicles, the tangent of the tilt table angle can be used to estimate the lateral acceleration, in g, necessary for the vehicle to reach the point of roll instability for steady state inputs. For this test, an M1077 flatrack was utilized. The 18 ton payload was switched to an 18 ton ISO container with a 24 inch high CG for the remainder of the EOA, AOA and performance testing.

Table 25
Tilt Table Instrumentation

Channel	Location & Orientation	Units
1	Front bumper inclinometer	deg
2	Front axle inclinometer	deg
3	Rear axle inclinometer	deg
4	Front axle inclinometer	deg
5	Tilt table inclinometer	deg

14.2 Tilt Table Results

14.2.1 LVSR-TD Curb Weight, Cross-Country Tire Pressure

While tilting with the passenger side of the vehicle toward the downslope side representative of making a left turn, the vehicle lifted the fifth axle onto the restraint straps at an average tilt table angle of 29.1 degrees. While tilting with the driver side of the vehicle toward the downslope side, representative of making a right turn, the vehicle lifted the fifth axle onto the restraint straps at an average tilt table angle of 23.7 degrees. The tilt angles for each tilt are provided in Table 26.

14.2.2 Curb Weight, Highway Tire Pressure

While tilting with the passenger side of the vehicle toward the downslope side representative of making a left turn, the vehicle lifted the fifth axle onto the restraint straps at an average tilt table angle of 28.6 degrees. While tilting with the driver side of the vehicle toward the downslope side, representative of making a right turn, the vehicle lifted the fifth axle onto the restraint straps at an average tilt table angle of 24.7 degrees. The tilt angles for each tilt are provided in Table 27.

14.2.3 Payloaded, Cross-Country Tire Pressure

While tilting with the passenger side of the vehicle toward the downslope side, representative of making a left turn, the vehicle reached the point of instability and lifted into the restraint straps at an average tilt table angle of 25.6 degrees. While tilting with the driver side of the vehicle toward the downslope side, representative of making a right turn, the vehicle reached the point of instability and lifted into the restraint straps at an average tilt table angle of 26.6 degrees. The tilt angles for each tilt are provided in Table 28.

14.2.4 Payloaded, Highway Tire Pressure

While tilting with the passenger side of the vehicle toward the downslope side, representative of making a left turn, the vehicle reached the point of instability and lifted into the restraint straps at an average tilt table angle of 23.7 degrees. While tilting with the driver side of the vehicle toward the downslope side, representative of making a right turn, the vehicle reached the point of instability and lifted into the restraint straps at an average tilt table angle of 26.1 degrees. The tilt angles for each tilt are provided in Table 29.

Table 26
LVSR-TD, Curb Weight, Cross-Country Tire Pressure
Tilt Angles At Point Of Instability
And Frame Twist Angles

Tilt Angle (degrees)	**Passenger Side Down**			**Driver Side Down**		
	Tilt 1	Tilt 2	Tilt 3	Tilt 1	Tilt 2	Tilt 3
Tilt Table Angle	28.6	29.4	29.3	24.2	23.4	23.5
Front Bumper Angle	32.7	33.3	33.0	28.7	27.3	27.4
Axle #1 Angle	32.6	33.2	33.1	28.1	26.8	27.0
Rear Bumper Angle	31.1	31.8	31.6	26.2	24.9	25.2
Axle #5 Angle	32.4	33.1	32.9	26.6	25.4	25.5
Front Bumper to Rear Bumper Twist	1.6	1.5	1.4	2.5	2.4	2.2

Table 27
LVSR-TD, Curb Weight, Highway Tire Pressure
Tilt Angles At Point Of Instability
And Frame Twist Angles

Tilt Angle (degrees)	**Passenger Side Down**			**Driver Side Down**		
	Tilt 1	Tilt 2	Tilt 3	Tilt 1	Tilt 2	Tilt 3
Tilt Table Angle	27.9	28.9	29.0	24.1	25.0	25.0
Front Bumper Angle	31.4	31.9	32.0	28.1	28.4	28.5
Axle #1 Angle	31.2	31.7	31.8	27.3	28.0	28.0
Rear Bumper Angle	29.8	30.3	30.6	25.8	26.4	26.1
Axle #5 Angle	30.9	31.9	32.0	26.2	26.7	26.7
Front Bumper to Rear Bumper Twist	1.6	1.6	1.4	2.3	2.0	2.4

Table 28
LVSR-TD, Payloaded, Cross-Country Pressure
Tilt Angles At Point Of Instability
And Frame Twist Angles

Tilt Angle (degrees)	**Passenger Side Down**		**Driver Side Down**		
	Tilt 1	Tilt 2	Tilt 1	Tilt 2	Tilt 3
Tilt Table Angle	25.3	25.8	26.3	26.7	26.7
Front Bumper Angle	30.3	30.8	32.3	32.6	32.6
Axle #1 Angle	30.3	31.0	31.5	31.8	31.8
Rear Bumper Angle	30.4	30.9	31.6	31.6	31.8
Axle #5 Angle	29.0	29.7	30.7	31.1	31.2
Front Bumper to Rear Bumper Twist	0.1	0.1	0.7	1.0	0.8

Table 29
LVSR-TD, Payloaded, Highway Pressure
Tilt Angles At Point Of Instability
And Frame Twist Angles

Tilt Angle (degrees)	**Passenger Side Down**			**Driver Side Down**		
	Tilt 1	Tilt 2	Tilt 3	Tilt 1	Tilt 2	Tilt 3
Tilt Table Angle	23.8	23.6	23.6	26.1	26.1	26.1
Front Bumper Angle	29.0	29.5	29.4	32.3	32.2	32.3
Axle #1 Angle	29.2	29.6	29.4	31.4	31.2	31.1
Rear Bumper Angle	28.3	29.3	29.0	30.8	30.9	31.0
Axle #5 Angle	28.9	29.6	29.3	30.6	30.6	30.4
Front Bumper to Rear Bumper Twist	0.7	0.2	0.4	1.5	1.3	1.3

The point of static instability of the vehicle on the tilt table was determined when at least one of the vehicle’s wheels lifted off of the surface of the table and the vehicle was restrained from rolling off of the tilt table.

The lateral acceleration necessary for rollover, or "rollover threshold," can be calculated by taking the tangent of the average tilt table angles at the point of instability. The right turn rollover threshold, i.e. tan(Ø), was calculated to be 0.44 g for the curb weight, cross-country tire pressure configuration. The left turn rollover threshold was calculated to be 0.56 g. The right turn rollover threshold was 0.46 g for the curb weight, highway tire pressure configuration. The left turn rollover threshold was 0.55 g. The right turn rollover threshold was 0.50 g for the payloaded weight, cross-country tire pressure configuration. The left turn rollover threshold was 0.48 g. The right turn rollover threshold was 0.49 g for the payloaded weight, highway tire pressure configuration. The left turn rollover threshold was 0.44 g.

The lateral acceleration necessary for rollover may be used to approximate the maximum speed that the vehicle could travel around a level, constant radius, steady state, non-vibratory turn without rolling over. Using the following calculation, the speed through a constant radius turn necessary for a rollover may be approximated: (Reference Table 30 for these results).

$$\text{Acceleration [g]} = \frac{0.0668 * \text{Velocity[mph]}^2}{\text{Turn Radius [ft]}}$$

The axial, lateral and vertical centers of gravity (CG) were calculated using the mass moment method. This is defined as:

$$CG = y_{CG} = \frac{\sum wy}{\sum w}$$

Where:
w = Weight of corresponding axle in pounds
y = distance of corresponding weight with respect to a fixed reference plane

The average CG location, calculated from CG values found for each tilt, is reported in Table 30.

The lateral center of gravity for the curb weight configuration is 0.5 inch toward the driver side of the vehicle centerline. It is 1.2 inches toward the driver's side of the vehicle for the fully payloaded configuration.

The longitudinal center of gravity for the curb weight configuration is 135.4 inches aft of axle #1. It is 183.4 inches aft of axle #1 for the fully payloaded configuration.

The average vertical center of gravity for the curb weight, cross-country tire pressure configuration is 66.0 inches off the ground. For the curb weight, highway tire pressure configuration, the average vertical center of gravity is 65.0 inches off the ground. For the fully payloaded, cross-country tire pressure configuration, the average vertical center of gravity is 66.3 inches off the ground. For the fully payloaded, highway tire pressure configuration, the average vertical center of gravity is 70.0 inches off the ground (Reference Table 30).

Table 30
LVSR-TD, Rollover Threshold, Vertical Center of Gravity, and Estimated Maximum Velocity 200-Foot Radius Steady State Turn

Tilt	Tilt Table Angle Ø (Degrees)	Rollover Threshold (Lat. g)	Vertical Center of Gravity	Est. Maximum Velocity (MPH)
Passenger Side Down, Curb, Cross-Country Pressure	29.1	0.56	58.2	40.8
Driver Side Down, Curb, Cross-Country Pressure	23.7	0.43	73.8	36.3
Passenger Side Down, Curb, Highway Pressure	28.6	0.54	59.5	40.4
Driver Side Down, Curb, Highway Pressure	24.7	0.46	70.4	37.1
Passenger Side Down, Payloaded, Cross-Country Pressure	25.5	0.47	67.7	37.8
Driver Side Down, Payloaded, Cross-Country Pressure	26.6	0.50	64.8	38.7
Passenger Side Down, Payloaded, Highway Pressure	23.6	0.44	73.9	36.2
Driver Side Down, Payloaded, Highway Pressure	26.1	0.49	66.1	38.3

14.3 Handling

Handling tests were performed to determine the dynamic safety limits of a vehicle. The dynamic reaction of a vehicle that results from sudden extreme driver inputs is directly related to safety. Such extreme inputs could ultimately lead to instability and loss of control of a vehicle. This data was also used to verify the computer model's performance near the limits of handling.

14.3.1 Theory of Handling Tests

Figure 9 provides a physical representation of wheel slip angle and the double lane change maneuver.

Depending on several parameters such as tire characteristics, shock damping rates, spring rates, bushing properties, chassis stiffness, location of drive wheels, centers of gravity, etc., a vehicle may experience conditions of oversteer or understeer. Understeer is commonly referred to as "push" and occurs when the front wheel slip angle is greater than the rear wheel slip angle. Oversteer satisfies the opposite conditions. Both can result in loss of control. However, understeer is considered the preferred condition for handling purposes. The dynamics of a vehicle which experiences understeer are less severe than those of oversteer. This is because understeer results in straight-line travel and minimal yawing effects are present. Considerable yawing is more commonly associated with oversteer conditions and can result in complete loss of control.

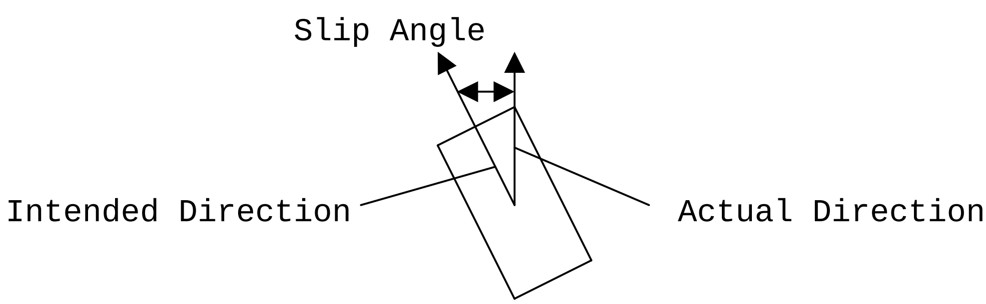


Figure 9
Diagram of Wheel Slip Angle, Top View

A condition more often experienced by larger vehicles is roll. It is common for larger vehicles to become unstable and roll over before the tires slip enough to cause any considerable amount of understeer or oversteer. The rollover tendency of large vehicles can be explained by a high center of gravity, as well as the large contact patches of the tires used on such vehicles. It should be noted however, that conditions of oversteer or understeer can be present during rolling conditions. A common goal of the 200-foot radius test is to determine the maximum speed and lateral acceleration a vehicle can withstand (commonly referred to as the vehicle's "end limit") before it rolls over or becomes uncontrollable due to sliding.

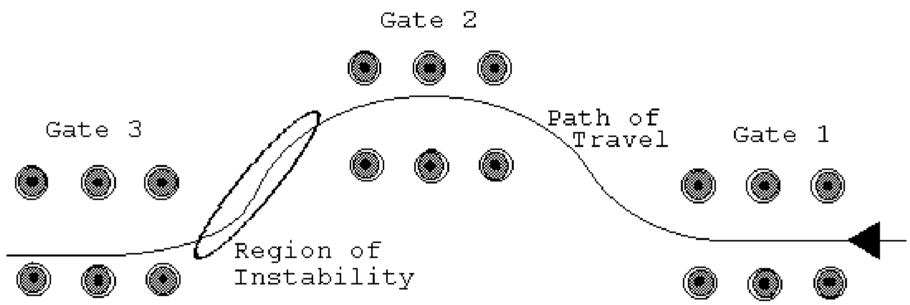


Figure 10
Diagram of Double Lane Change Maneuver

The double lane change highway type maneuver tests the vehicle's dynamic characteristics under conditions of sudden steering input and vehicle weight transfer (Figure 10). Significant body roll and yawing are often a common effect on large vehicles during this test, especially transitioning from the second to third gates.

14.3.2 Handling Instrumentation

Prior to conducting the dynamic handling engineering test, the vehicle was instrumented to measure parameters as in Tables 31 and 32:

Table 31
LVS Handling Instrumentation

Channel	Location and Orientation	Units
1	Speed	mph
2	#1 axle lateral accelerometer	g
3	#4 axle lateral accelerometer	g
4	Chassis C.G. lateral accelerometer	g
5	Pitman arm steer angle	deg
6	#1 axle left wheel angle	deg
7	#1 axle right wheel angle	deg
8	Roll rate at center of gravity	deg/s
9	Yaw rate at center of gravity	deg/s
10	Left articulation cylinder displacement	inch
11	Right articulation cylinder displacement	inch

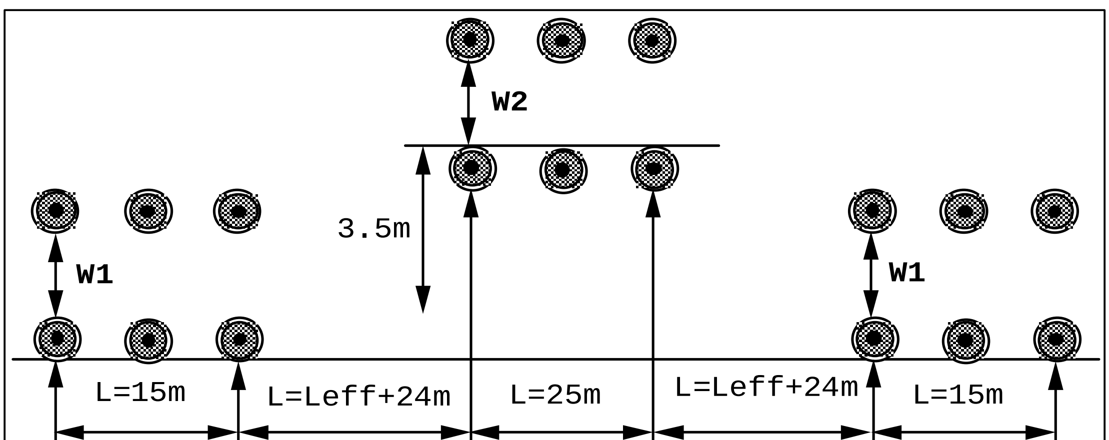
Table 32
LVSR Variants Handling and Ride Quality Instrumentation

Channel	Location and Orientation	Units
1	Driver seat base vertical accelerometer	g
2	Driver seat base lateral accelerometer	g
3	Driver seat base longitudinal accelerometer	g
4	#1 axle left side vertical accelerometer	g
5	#3 axle left side vertical accelerometer	g
6	#5 axle left side vertical accelerometer	g
7	Frame at #1&#2 center vertical accelerometer	g
8	Frame at #4 axle left side vertical accelerometer	g
9	#1 axle left side lateral accelerometer	g
10	#3 axle left side lateral accelerometer	g
11	#5 axle left side lateral accelerometer	g
12	CG lateral accelerometer	g
13	CG inclinometer (Roll)	deg
14	CG gyro (Yaw rate)	deg/s
15	#1 axle wheel travel	in
16	#3 axle wheel travel	in
17	#5 axle wheel travel	in
18	Steering wheel angle	deg
19	#1 axle left wheel angle	deg
20	#3 axle left wheel angle	deg
21	#5 axle left wheel angle	deg
22	Speed	mph

14.3.3 Handling Test Procedures

Handling tests were performed using the 200-foot constant radius and the double lane change. The 200-footconstant radius test was performed in both directions at variably increasing speeds. A qualified driver operated the vehicle, while engineers and technicians observed the vehicle’s behavior. These maneuvers were implemented on a high coefficient smooth, dry surface (asphalt) at increasing speed increments. The vehicles were tested at different speed ranges in both the clockwise (CW) and counter-clockwise (CCW) directions. The course surface was flat to eliminate the road crown as a variable. The course was delineated with a centerline marker for driver tracking.

The double lane change test was performed with the use of three sections of cones, each representing the required “gate” or path that the vehicle was required to enter and proceed through. The cone layout and test procedure is specified in North Atlantic Treaty Organization (NATO) Allied Vehicle Testing Publications (AVTP) 03-160 and SAE J2014 (Figure 11). The test location was a smooth flat paved surface. The test was conducted at increasing speeds up to the end-limit of the vehicle (Reference Photograph Number 20238-26).



W1 = 1.1(VW) + 0.25m

W2 = 1.2(VW) + 0.25m

VW = Vehicle Width

Leff = Overall Length of the Vehicle Measured at 0.5m from the Ground

Figure 11
Double Lane Change Configuration

14.4 Handling Results

14.4.1 200-Foot Constant Radius – LVSR Variants

Tables 33 through 38 show the maximum speeds that wheel lift at the number 5 axle first occurred for the 200-foot constant radius curve. The maximum lateral acceleration at the point of wheel lift of the number 5 axle is shown in Tables 33 through 38 for each vehicle configuration. The criteria for near end limit for the physical testing was to slowly increase the speed on the 200-foot constant radius event until wheel lift occurred at the #5 axle only. This was well below the point of dynamic instability. Once the model was validated with the constant radius data, the modeling and simulation environment was used to calculate the end limit.

The data is repeated for both clockwise and counterclockwise travel. In all cases, the target speed was 34.6 MPH, which corresponds to a 0.4 g lateral acceleration at the CG for this steady state curvature.

Table 33
200-Foot Constant Radius – LVSR-TD
18 Ton CCGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
CCW	31.5	0.40
CW	29.8	0.37

Table 34
200-Foot Constant Radius – LVSR-TD
22 Ton HGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
CCW	29.6	0.28
CW	29.5	0.37

Table 35
200-Foot Constant Radius – LVSR-TD FPU/Rayden RBU
18 Ton CCGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
CCW	29.7	0.31
CW	28.6	0.37

Table 36
200-Foot Constant Radius – LVSR-TD FPU/Rayden RBU
22 Ton HGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
CCW	29.8	0.31
CW	29.0	0.28

Table 37
200-Foot Constant Radius – LVSR-TD FPU/Hendrickson RBU
18 Ton CCGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
CCW	29.6	0.33
CW	28.6	0.28

Table 38
200-Foot Constant Radius – LVSR-TD FPU/Hendrickson RBU
22 Ton HGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
CCW	29.2	0.27
CW	28.2	0.31

Again, using the computer models validated with the above instrumentation, the LVSR-TD was optimized in the model and run over the same constant radius event. The suspension and roll control mechanism (i.e., roll bar) were optimized to produce the best combination of empty and loaded ride quality, empty and loaded shock attenuation (10 inch half-round) and stability events. With the suspension optimization and steering at the number 5 axle for the Raydan and Hendrickson RBU configurations, all variants were able to exceed the 0.4 g lateral acceleration threshold at the CG or 34.6 MPH in a 200-foot constant radius curve before the point of dynamic instability.

14.4.2 Double Lane Change – LVSR Variants

Tables 39 through 44 show the maximum speeds that wheel lift first occurred at the number 5 axle for the NATO AVTP 03-160 double lane change event. The maximum lateral acceleration at the number 5 axle is shown in Tables 39 through 44 for each vehicle configuration. The criteria for near end limit for the physical lane change testing was to slowly increase the speed through the event until wheel lift occurred at the number 5 axle. The lane change was then repeated at that speed until the lane change was negotiated without hitting the cones marking the path of the lane change. Again, this was below the point of dynamic instability. Once the model was validated with the double lane change data, the modeling and simulation environment was used to calculate the end limit.

The data is repeated for both directions of travel. In all cases, the target speed was 45 MPH, in accordance with the draft Performance Specification requirements.

Table 39
Double Lane Change – LVSR-TD
18 Ton CCGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
W to E	34.6	+0.43 -0.43
E to W	35.4	+0.35 -0.39

Table 40
Double Lane Change – LVSR-TD
22 Ton HGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
W to E	39.1	+0.37 -0.28

Table 41
Double Lane Change – LVSR-TD FPU/Raydan RBU
18 Ton CCGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
E to W	31.7	+.30 -.35

Table 42
Double Lane Change – LVSR-TD FPU/Raydan RBU
22 Ton HGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
E to W	33.6	+.32 -.39

Table 43
Double Lane Change – LVSR-TD FPU/Hendrickson RBU
18 Ton CCGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
E to W	39.5	+0.50 -0.43
W to E	38.2	+0.38 -0.28

Table 44
Double Lane Change – LVSR-TD FPU/Hendrickson RBU
22 Ton HGVW and Paved CTIS Setting

Turn Direction	Speed (mph)	Measured Lateral Acceleration #5 Axle (g)
E to W	36.8	+0.43 -0.35

Again, using the computer models validated with the above instrumentation, the LVSR-TD was optimized in the model and run over the same double lane change event. The suspension and roll control mechanism (i.e., roll bar) were optimized to produce the best combination of empty and loaded ride quality, empty and loaded shock attenuation (10 inch half-round) and stability events. With the suspension optimization and steering at the number 5 axle for the Raydan and Hendrickson RBU configurations, the LVSR-TD was able to meet the 45 MPH in NATO AVTP 03-160 double lane change event at HGVW. The Hendrickson hydraulic suspension RBU was able to negotiate the lane change at 45 MPH but the results were questionable. Given eight equally spaced cones per gate, the vehicle hit four cones in Gate 2 and four cones in Gate 3. The Raydan air ride RBU simulation predicted rollover during the transition to the second gate.

14.5 Stability and Handling Conclusions

Given a combination of physical testing and modeling and simulation with a validated model, the LVSR-TD met the draft Performance Specification requirements for the constant radius and NATO AVTP 03-160 double lane change events. The Hendrickson hydraulic RBU was questionable and the Raydan air ride did not meet the requirements.